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THE DESIGN OF
STEAM BOILERS
AND
PRESSURE VESSELS

BY

GEORGE B. HAVEN, S.B.
Professor of Machine Design, Massachusetts Institute of Technology

AND

GEORGE W. SWETT, S.B.
Assistant Professor of Machine Design, Massachusetts Institute of Technology

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PREFACE.

THE design of machinery is sometimes defined as the science of reasonable approximation. Comparatively few of the methods necessary in fixing the proportions of machine parts fall under the established formulæ of Physics or Applied Mechanics. It is only by a skilful adaptation of theoretical deductions to practical requirements that the science of Machine Design may claim a right to legitimate and useful existence.

With the hope of harmonizing still further the rational methods of both theory and practice, the authors of this volume have sought to incorporate in its pages the most direct procedure possible in the design of pressure apparatus. As a basis for the broader study of Machine Design it is believed that problems of this character possess peculiar advantages. The definiteness of the loads in pressure vessels and the reliability of the materials usually employed in their construction are incentives to intelligent and accurate calculation. Wherever possible, results have been obtained by rational rather than empirical methods. The usages of current boiler-making practice have been kept constantly in view. Numerous assumptions have necessarily been made but in every case it is believed a conservative reason has been given as their basis.

The last five chapters serve to illustrate the application of the principles and formulæ previously deduced to the practical design of various types of boilers and pressure vessels. It is hoped that the definite program followed in these problems may be of value not alone to students but to the boiler-making profession as well. The data and constants assumed are derived from actual practice and represent wide margins of safety in regard to boiler performance and construction. The design of water tube boilers is largely the result of ripe experience and well established precedent. Aside from a few features in connection with the steam drum, their proportions can rarely be determined by calculation. Hence it has not been considered wise to include in this work the specific design of a boiler of this class.

The authors wish to acknowledge their sincere indebtedness to Professor Peter Schwamb with whom their long association at the Massachusetts Institute of Technology will always be an inspiring remembrance. Much of the subject matter of this treatise is the direct result of his guidance and suggestion.

The work on "Steam Boilers" by Professors Peabody and Miller has been of much assistance in the preparation of this volume and the authors are under deep obligation to them as well for kindly criticism and advice.

The gratitude of the authors is due Mr. J. W. F. Macdonald, chief draughtsman of The International Engineering Works, for many practical suggestions.

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G. B. H.
G. W. S.

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THE DESIGN OF STEAM BOILERS AND PRESSURE VESSELS

CHAPTER I.

GENERAL PRINCIPLES.

BEFORE the construction of pressure apparatus of any kind can be undertaken a clear conception must be gained of the fundamental principles which underlie such work. The design of steam boilers entails much more than an extended application of the theory of mechanics or of the strength of materials. The presence of heat energy and the effect of impure water complicate the problem and introduce many questions outside the usual realm of machine design.

The vast increase in the use of steam-generating apparatus during the last fifty years has been the natural accompaniment of the corresponding growth in manufactures and commerce. Considering the last three decades alone, the United States census reports show that the number of horse-power developed by steam boilers connected with manufacturing operations has increased from approximately 2,000,000 in 1880 to 14,000,000 in 1910. When it is considered that these figures represent conditions in this country alone, and comprise but a small fraction of the total amount of steam and hot water handling machinery, the importance of sound and rational methods of design for such apparatus needs no further emphasis.

The appalling catastrophies which, from time to time, fill the columns of the public press indicate in no uncertain manner the fearful hazard to which multitudes of human lives are daily subjected by proximity to confined steam pressure. The heating of buildings, the transportation of passengers by rail and water, and the generation of power for manufacturing purposes bring into all walks of life the presence of vast aggregations of energy, the handling of which must be accomplished with consummate care or fearful casualties will result. The very first requisite, therefore, to be considered in the design of steam boilers and other pressure vessels is safety. No one can measure the damage to life and property which exploding pressure apparatus may entail. Operating efficiency and commercial economy fall into insignificance when compared with safety. That steam handling appliances should confine their working pressures with a liberal margin of safety must always therefore be made the basis of the calculation and design of such apparatus.

1. Difficulties in Design. — The increasing difficulty in the design and manufacture of steam boilers springs from a number of sources. With the use of steam turbines and reciprocating engines of multiple stage and expansion there has been a great increase in the steam pressures required. Whereas a few years ago pressures of 100 to 125 lbs. per square inch were considered high, at present steam apparatus is often designed to sustain from 200 to 250 lbs. per square inch.

The number of horse-power comprised in one boiler unit has also been greatly increased of late. A one-hundred horse-power boiler was once considered a good-sized power unit. Most manufacturers at present guarantee a development of two to three times this amount per boiler and in case of water-tube boilers, horse-power units ranging as high as 600 and 700 are frequently encountered.

The congested condition of manufacturing enterprises in cities, as well as the increased horse-power added to old plants as they develop, often puts floor space at a premium. This requires vast aggregations of power in very limited quarters and frequently necessitates the placing of boilers in tiers upon successive floors. The complications arising from such arrangements call for very great skill and foresight on the part of the designer.

With the increased cost of fuel there has arisen a demand for the generation of steam by burning various grades of coal formerly regarded as refuse. As an instance of this, the Lackawanna Railroad some time ago adopted a type of locomotive with wide grate especially designed for burning culm or anthracite slack. In this manner accumulations of fuel previously considered as useless have been made to serve a valuable purpose and return a generous revenue to the railroad. The construction of such very wide fire boxes has proved a difficult problem for the designer.

The purity of the water used in generating steam has much to do with the life and efficiency of the boiler. Especially is this true in the Western States, where many waters are found heavily impregnated with the carbonates of lime and magnesia. The precipitations from such feed waters after boiling are very considerable. When allowed to settle and burn on, such deposits form a thick scale which interferes seriously with the prompt transfer of heat through the boiler plate. For this reason where such feed waters are to be used the designer must allow wide margins of safety to compensate for the weakening effect of overheated and burned plates.

With the development of the steam turbine and the constant demand for stricter economy in the consumption of steam, the use of the superheater has come into prominence. For the lower degrees of superheating a portion of the boiler itself may be made to comprise the superheater. Such, for instance, is the function of the fire tubes of vertical boilers where they pass through the steam space. For high degrees of superheating a separate piece of apparatus placed in the path of the hot gases, or over a furnace entirely independent of the boiler, may be made to perform the office. In either event the transfer of heat to the vapor is not as prompt as to a liquid. Superheating members must therefore be designed with direct reference to their wasting away under the effects of overheating and with full provision for their repair and renewal.

The last and perhaps the most serious of the difficulties to be encountered in designing boilers is the necessity of securing a large capacity for overload. Time was, when the manufacturer of boilers considered that he had done extremely well in providing for an overload of one-third the original rating. At present, however, under certain conditions in naval vessels, as well as in the generation of current for use in electric traction, provision must be made for carrying peak loads two to three times the normal horse-power. By means of certain mechanical stokers, boilers have been forced to these excessive capacities for very short intervals. This is not done by an increase in pressure but rather by a more rapid coal consumption under forced draft. To endure such treatment without injury and insure the safety of operatives and attendants as well, the boiler must embody factors of safety in its design upon which entire reliance may be placed at all times.

2. Recent Improvements. — In contrast to the above difficulties in the design of modern pressure vessels, there may be enumerated certain great advances. Far better materials than formerly await the hand of the designer and manufacturer. Comparatively few years ago steel was not considered a fit material for boiler shells, because of its lack of ductility. Wrought iron was used exclusively for the purpose. At present, however, " flange steel " is obtainable of a tensile strength one-fourth greater than that of wrought iron, and at the same time with sufficient ductility for the most exacting requirements. The same is true of rivet steel. But very few designers now specify puddled wrought iron for boiler rivets. It is a significant fact in this connection that in 1909 one of the largest manufacturers of boiler tubes in the United States voluntarily relinquished the use of wrought iron for the purpose, devoting since then their entire energies to the manipulation of soft steel for pipes and tubes. It would appear from this that modern open-hearth steel is soon to take its place as the material preëminently fitted for the manufacture of steam boilers throughout.

There is a second great advance as regards machine tools. The modern boiler maker has at his command powerful and accurate tools for performing every class of operation. Handwork, with its personal equation and consequent lack of uniformity, has largely been eliminated while accurate and uniform mechanical operations have taken its place. Portable tools driven by air or electricity relieve the mechanic of many arduous tasks and provide him with a never failing source of energy. A boiler shop is now conducted with much the same definiteness of organization that one will find in a first-rate machine shop. Slipshod methods have given place to intelligence and directness. The drift-pin, once a much used tool in riveted work, has largely been retired to the shelf.

Lastly may be mentioned the ever-increasing array of laws and ordinances which have been framed of late to keep the boiler maker's feet in paths of rectitude. Federal supervision over the boilers of steam vessels is ample and well administered.

The principles laid down in the " Rules of Supervising Inspectors " are sound and conservative. Among the states, Massachusetts has taken the lead in making the design and manufacture of boilers and air tanks a matter of public statute and in placing the inspection of such pressure vessels in the hands of the State Police. Other states and countries, to a more or less degree, have followed in the same direction. The Dominion of Canada has an excellent set of boiler rules. Besides these legal enactments many private insurance and casualty companies have in force excellent codes of regulation.

3. Boiler Horse-power. — The use of the term horse-power in connection with steam generators is at best more or less arbitrary. Originally the term was intended to convey the idea that, with the given boiler attached to an engine of ordinary economy and developing under rather adverse conditions the stated horse-power, there was an ample margin of evaporative resource on the part of the boiler.

In the year 1885 this rating was summarized by the American Society of Mechanical Engineers in the statement that one boiler horse-power was equivalent to the evaporative energy necessary to turn 30 pounds of feed water per hour taken into the boiler at 100° F. to dry steam at 70 pounds per square inch gage pressure. Since the stereotyped conditions of the above definition were rarely found to fit a given case in practice, the same organization of engineers in 1898 defined a boiler horse power as equivalent to the evaporation of 34.5 pounds of water per hour to dry steam from and at a temperature of 212° F. This was practically equivalent to the performance given under the definition of 1885. The heat of vaporization of one pound of dry steam at 212° F. is 969.7 B.T.U. Hence one boiler horse-power corresponds to the absorption of 33,455 B.T.U. per hour.

There is no definite relationship between boiler and engine horse-power. Depending upon the excellence of the valve gear and general design, as well as upon the pressure and degree of saturation of the steam used, an engine will develop one indicated horse-power upon a water consumption varying from 10 to 60 pounds per hour. Hence one boiler horse-power roughly corresponds to a range of engine horse-powers varying from 3 to 0.5.

A more accurate comparison may be made with reference to the heat consumption of the engine. A modern steam engine of approved pattern will generate one indicated horse-power at an expenditure varying from 200 to 400 B.T.U. per minute. The heat consumption depends upon whether the engine is arranged for single or multiple expansion, for saturated or superheated steam, or for running condensing or non-condensing. Taking the thermodynamic equivalent of a boiler horse-power as 33,455 B.T.U. per hour and the above range of heat consumption at the engine, one boiler horse-power will cover from 1.4 to 2.8 engine horse-power. Wherefore in designing large plants it is generally considered sufficient to provide boiler horse-power equal to half the indicated horse-power of the engines, there being at the same time sufficient margin for laying off boilers due to cleaning and repairs.

In framing the above general definition the committee of 1898 made the following statement: "A boiler rated at any stated capacity should develop that capacity when using the best coal ordinarily sold in the market where the boiler is located, when fired by an ordinary fireman, without forcing the fires, while exhibiting good economy; and, further, the boiler should develop at least one-third more than the stated capacity when using the same fuel and operated by the same fireman, the full draft being employed and the fires being crowded, the available draft in the flue just beyond the boiler, unless otherwise understood, being not less than one-half inch water column."

4. Thermal Boiler Efficiency. — The relation of the heat absorbed by the water in the boiler to that generated by the combustion of the fuel is the efficiency ratio for a given boiler performance. With good firing and a well-designed plant this figure ranges from 65 to 75 per cent.

5. Sizes of Units. — While there are no exact limits in regard to the maximum or minimum generating capacity for which a boiler unit may be designed, there are, however, certain practical considerations in relation to the type of boiler chosen which govern the sub-division of horse-power. Small boilers for special purposes have been designed without definite limit as to rating. For manufacturing enterprises it is rarely wise to install stationary boilers of less than 50 horse-power, although horizontal return tubular boilers are built in sizes as small as

20 horse-power. Portable and semi-portable boilers of the locomotive and vertical type are frequently designed for units as small as 15 horse-power. For Scotch boilers the question depends upon the accommodation of the internal furnace. To provide reasonable grate room in a combustion tube, requires a minimum diameter for the external shell of approximately 6 feet and a corresponding rating of about 50 horse-power. Sectional and water-tube boilers are in general concentrated power units of high rating. Except for steam-heating purposes it is rarely good practice to specify such boilers of less than 100 horse-power.

Passing to the other extreme, the concentration of power in very large units depends primarily upon the ability of metals to transmit heat while loaded with severe hoop tensions and compressions. A return tubular boiler 78 inches in diameter, using a quadruple riveted butt joint, requires a steel shell nine-sixteenths of an inch thick in order to preserve a reasonable factor of safety. This has been found to be about the maximum thickness for safe use in externally fired boilers. The rating of such a boiler by A.S.M.E. standards and under ordinary conditions is 200 horse-power and is therefore the largest practical unit for this type. With forced draft or a very high stack this figure would be considerably exceeded.

Locomotive boilers in railroad practice are not rated in the usual manner for boiler horse-power, since the conditions of their operation vary so widely from the normal. The indicated horse-power of large locomotives when measured at the engine cylinders has approximated 2000 for short intervals and under the severest conditions of forced working. It is probable, however, that the draft induced by the exhaust nozzle and the economy gained by the superheater provided for a very large percentage of overload so that the actual horse-power of the boiler would be reduced to about one-half of the above amount. Special stationary locomotive type boilers have been designed for 1000 horse-power but the forcing necessary to secure this figure does not conduce either to long life or a small repair bill. For economical working a limit of 150 horse-power may be set for semi-portable locomotive type boilers and about twice this figure for stationary ones.

Remarks of much the same tenor may be made in relation to

Scotch boilers. For land use the dry back Scotch boiler with two furnace tubes is limited to about 300 horse-power. In marine use with four furnace tubes and forced draft five times this figure is often reached.

Since in vertical boilers there are no thick plates directly exposed to the fire, the limit of horse-power is dependent upon the possibility of staying the furnace sheets and the durability of the superheating surface at the upper end of the tubes. Unfortunately the lower tube sheet in boilers of this type is located directly in the path of the falling sediment from the surface of the tubes above and is at the same time exposed to the maximum heat of the fire from below. Provision must therefore be made for the easy cleaning of such tube sheets if the life of the boiler is to be prolonged. The rating of vertical boilers is generally limited to units of 400 horse-power.

The water-tube or safety boiler excels all other types in the amount of horse-power developed per unit of its own weight. For stationary use it is also capable of greater concentration of power than other types. Units aggregating 500 horse-power are frequently used in practice and in some instances the size has been increased to 700 horse-power.

In selecting the size of units for a given plant it must be borne in mind that, while the efficiency of the boiler and the economy of its attendance and operation are generally higher the larger the unit, at the same time the inconvenience due to laying off for cleaning and repair is greatly increased. The unit chosen therefore should generally be so small a factor of the total capacity of the plant as to permit a good degree of flexibility.

6. Range of Pressure. — The pressure in most of the steam-generating apparatus used for heating buildings is not supposed to exceed 15 pounds per square inch. Such appliances if protected from over-pressure by devices of approved accuracy are exempt from the usual boiler rules and regulations. It is only when there is liability to severe overloading by accident or carelessness that such pressure vessels explode and cause serious damage.

With pressures ranging from 15 to 100 pounds per square inch there is a vast amount of low-pressure steam machinery in use in which the generation of power and the utilization of steam for industrial purposes are of paramount importance. Such pressures may be regarded as the conservative practice of some years ago.

For the ordinary generation of power in moderately large units, pressures ranging from 100 to 175 pounds per square inch are of common occurrence. The leakage of joints and gaskets at such pressures is not a very serious matter and can be prevented by the use of "extra heavy" pipe fittings.

Where the highest degree of economy is sought in large units operating under ideal conditions, steam pressures have of late been increased to the region of 250 pounds per square inch. Especially is this true in the case of modern locomotive and steam turbine practice. Most locomotives at present are designed for 225 pounds pressure. Such pressures are more or less troublesome to handle, requiring the greatest of care in making up and maintaining joints against leakage. Special boilers of peculiar design have been built for pressures considerably in excess of those just mentioned, but they belong to the realm of physical apparatus rather than to that of power generators.

While the limit of pressure adopted is primarily a question of securing economy in the prime mover to be used, the type of steam generator has much to do with the question as well. An externally-fired boiler of reasonable size for pressures in excess of 175 pounds per square inch requires a shell thickness too great for durability. Hence horizontal return tubular boilers are rarely designed for pressures exceeding 150 pounds per square inch. The allowable pressures in vertical and Scotch boilers, where the external shell is unlimited in thickness, are governed by the ability of the internal furnace walls to resist collapse. Two hundred pounds per square inch is approximately the limit in such cases.

For the extreme pressures mentioned above, it is necessary to pass in practice to the water-tube boiler. Here the total volume of water is very much less and is confined in smaller channels. The thickness of metal therefore, even for 250 pounds per square inch, need not be excessive.

7. Superheating Appliances. — In generators of high-pressure steam there is generally included the necessary apparatus for superheating. Aside from the low degrees of superheat naturally obtained where the fire tubes of vertical boilers pass through the steam space, there are two classes of appliances used for the purpose. First, the superheating member may be housed directly in the boiler setting, using the heat from the ordinary fire. The alternative arrangement consists of placing the superheater over an independent furnace with grate and uptake entirely separate.

The form of the superheating vessel itself may consist of U-shaped tubes expanded at their free ends into steel headers and so arranged that a current of steam drawn through them will be far enough removed from contact with its liquid to become more or less superheated. Various other forms and devices, such as straight tubes placed one within the other and connected to separate chambers at their ends, have been used for the purpose of superheating steam.

Attached superheaters may be suspended in the natural path of the hot gases on their way to the uptake. Or again, a portion of the products of combustion may be by-passed directly from the main furnace to a separate chamber in which the apparatus is installed. In the former case the superheater is exposed to the full heat of the fire while the water in the boiler is approaching the point of evaporation. Unless a positive current of steam is maintained through such appliances they suffer immediate injury by overheating. Hence superheaters of this kind must be placed below the water line of the boiler and provision be made for filling or " flooding " them with water while raising steam. This expedient should be fully anticipated in the design of the apparatus. Means must be provided for the free circulation of water and the delivery of steam to the main boiler. At the same time the sediment precipitated in the superheater by the boiling of the water should be within easy reach of handholes and cleaning appliances. After reaching full steam pressure the flooding water in the superheater is discharged and a current of steam immediately established in its place. In shutting down boilers having this arrangement the same expedient is adopted to preserve the superheater from injury.

When the superheating chambers are connected by damper by-pass with the main furnace, the apparatus may be placed above the water line and designed to convey steam only. In raising steam this chamber is temporarily cut out of the path of the hot gases. At full steam pressure the saturated steam may be drawn through the superheater and the by-pass dampers opened.

Separately-fired superheaters should be arranged so that the full heat of the fire is broken and diffused by a reverberatory arch. The mass of brickwork thus provided stores an immense amount of heat, improves the combustion of the gases, steadies the operation of firing and prolongs the life of the superheating tubes. To increase the metal surface in contact with the products of combustion, cast-iron rings with projecting fins are strung on the outside of the steam conduits in certain forms of superheaters.

In designing such apparatus it must continually be kept in mind that the absorption of heat by a vapor is never so prompt as by a liquid. Overheated and burned metal is often the result therefore unless reliable means are provided for maintaining a rapid flow of steam through such appliances. All the members of superheaters are exposed to widely varying temperatures and full provision must be made in their design for the stresses caused by expansion and contraction.

8. Fuel, Grates and Settings. — The fuel, for the use of which the boiler is intended, has a considerable effect upon its design. All fuels require a large amount of excess space for the thorough mingling of the gases during combustion. Especially is this true of bituminous and other long-flaming coals. A short-circuit from grate to uptake tends toward large losses of heat up the chimney. Horizontal and water-tube boilers utilize the space behind the bridge wall for the purpose of combustion. Scotch boilers are usually accompanied by roomy combustion chambers either internal or external to the shell. Locomotive and vertical boilers are more or less lacking in this respect, but the high fire boxes usually found in such boilers compensate to a certain degree for the lack of adequate combustion space.

It is generally good practice, where manufacturing refuse is to be burned, to place the grate in a separate chamber outside the limits of the boiler setting. Such an arrangement, usually called a Dutch oven, provides sufficient brickwork to serve as a reservoir of heat in firing fresh or damp fuel and at the same time lengthens the gas passages so that a more perfect heat absorption takes place.

The criticism is often made that the furnace walls of internally-fired boilers never reach a temperature much in excess of that due to the accompanying steam pressure. Such comparatively cold walls do not, it is true, minister to efficient combustion but the absence of other heat losses more than counterbalances this disadvantage.

The spaces between the grate bars must vary with the size of fuel and the depth of fire to be carried. The air inlets through the grates are usually designed to comprise not less than forty per cent of the total area.

Ideal combustion consists of complete oxidation of all the elements contained in the coal. Sufficient air for this purpose should be introduced through the grate. It is never possible to admit the air theoretically necessary for the chemical action alone, since the access of the draft to all portions of the fire is not perfect. The partial oxidation of the coal constituents results in a serious loss of efficiency in the boiler performance. It has been found by chemical analyses of flue gases, as well as by general tests upon boiler plants, that from fifty to one hundred per cent of air for dilution must be provided in excess of the calculated amount necessary.

A large over-supply of air on the other hand robs the combustion of heat units and lowers the furnace temperature. For this reason leaky boiler settings conduce to poor economy and general inefficiency. The marked saving in fuel effected by internally-fired boilers, especially those of the vertical, Scotch and locomotive type, may be attributed in large measure to the fact that there is no possibility of air leakage through the setting. All the air admitted to the fire must pass through the grate with the exception of a small portion purposely introduced through the grid in the fire door or through perforations in the bridge wall. The settings for all types of externally-fired boilers, especially those of the water-tube class, are complex and present many opportunities for leakage through cracks and crannies as well as around door frames and cleaning holes. To obviate this difficulty most boiler manufacturers provide riveted sheet steel settings with an internal lining of brickwork. Such settings are very effective in saving fuel but are not widely used on account of their cost and rapid deterioration if the brick lining is not kept in good repair.

To prevent heat losses the walls of ordinary boiler settings should be at least four brick thick. Located at the center of the wall there should be an insulating air space at least two inches in width. Very often this air space is carelessly left unsealed and therefore, instead of serving as a means of insulation, becomes a duct for air leakage in communication with lining cracks.

The brickwork in the setting should never be laid rigidly against the surface of the boiler. If such is the case expansion and contraction will soon open cracks. Where a fire cut-off is to be maintained the brickwork should be kept back from one-half inch to one inch and the crack calked with asbestos rope. Expansion rollers must be provided under at least one set of supports to accommodate motion in a lengthwise direction.

When cast-iron or steel columns pass through the brickwork of boilers set singly or in battery, care should be taken that a space is left around such structural members for ventilation. This provides at the same time for renewal or removal of the column without disturbance to the setting and prevents injury to the column through overheating. The space thus provided should be connected by a duct not less than ten inches square with the external air to insure circulation.

Grate bars must never be tightly fitted between portions of the setting since their expansion will rack the brickwork. Either a free space should be provided at the end of the grate or the grate bearer should support the bars on an inclined surface up which they may slip without injury to the surrounding fixtures.

A flat bridge wall at the end of the grate keeps the fuel in place and serves as a support for the inner end of the grate. With the intense combustion necessary under forced draft, the bridge wall should be designed to contain a water cooled core through which is drawn the feed water. The temperature of the bridge wall is very much higher than that of the rest of the furnace walls. To prevent its expansion from heaving the adjacent lining, the bridge wall is often fitted at its ends into recesses in the side walls. The depth of these recesses is sufficient to accommodate the maximum expansion. The lower portion of the bridge wall is sometimes utilized for the introduction and distribution of air for dilution. Perforated bridge walls in Scotch boilers assist to a marked degree the narrow and low combustion space in the furnace tubes.

The opening behind bridge walls should not be filled in or leveled off but left an open chamber in which the gases may mix and eddy as their combustion proceeds. Feed and blow-off pipes passing through the gas passages should always be encased in sleeves of heavy cast iron pipe to prevent wasting upon their surfaces.

If the plant is dependent upon ordinary natural draft it is necessary that the gases be released at the uptake with a temperature of approximately 500° F. in order to maintain combustion. While this appears to be a large loss it is in a sense a necessary one. When special induced or forced draft apparatus is supplied, feed-water heaters or "economizers" may be used and the temperature of the chimney gases materially lowered. Such appliances, however, consume very considerable amounts of power in their operation, which should be charged to the plant when computing their economy.

In horizontal multitubular boilers the gases pass along the under side of the shell to the rear and return to the front by way of the tubes. The expedient has been tried of providing a third "pass" for the gases along the top of the boiler shell. This complicates the setting and brings heat to a portion of the shell likely to contain the riveted joints. It is not, therefore, a wise precedent to follow. At the rear of the boiler an arch of brickwork should protect the upper portion of the tube sheet, as well as the staying appurtenances, from the onrush of the flames.

Locomotive and Scotch boilers generally insure the mixing of gases in combustion by baffle walls or arches at the rear of the grate. A hanging baffle arch in furnace tubes is very effective in breaking up the current of gases and promoting thorough combustion. In locomotive fire boxes a baffle bridge wall performs the same function and protects the tube ends from overheating as well.

Vertical boilers are open to the criticism that the short direct transit of the flames from grate to uptake is conducive to excessive chimney temperatures. For this reason much smaller tubes may be used in such boilers and the velocity and temperature of the gases correspondingly reduced. At the same time a tall stack permits the use of an economizer for transmitting a large part of the chimney temperature to the feed water. When so arranged vertical boilers maintain high rates of economy while occupying a minimum of floor space.

Water-tube boilers are dependent for their heat-absorbing power upon many passes of the hot gases back and forth among the water tubes. In order to direct the transit of the flames in definite channels special forms of baffle tiling are inserted and clamped in place among the tubes. Experiments at the University of Illinois, Bulletin No. 34, show that a furnace roof, where the water tubes are completely encircled by the tiling, is but a few per cent less efficient than one in which the lower half of the tubes is directly exposed to the flames. In any event the form of tiling used should be such as to permit easy renewal without disturbing adjacent portions. The clamps which hold the blocks in place should be of the simplest and most inexpensive construction since iron subjected to such high temperatures wastes away very rapidly.

There are two general methods of directing the flow of gases back and forth among the tubes. In the first, the tiling is inserted in horizontal layers and the gases make three or more circuits in a direction parallel with the tubes. The second method employs tile partitions combined with a hanging bridge wall, and secures three or more passes in a direction perpendicular to the tubes. It is interesting to note in this connection that two tests performed under practically duplicate conditions by Mr. George H. Barrus, and reported in the *Engineering Record* of Feb. 19, 1898, indicate a boiler efficiency in the case of the horizontal tiling some six per cent in excess of that shown by the vertical tiling.

Whatever the form of boiler, the surfaces exposed to the fire must be readily accessible for cleaning. It is doubtful if engineers in general appreciate the resistance which moderately thick layers of soot offer to the absorption of heat. Soot itself is a first rate heat insulator, ranking some five times as efficient in this regard as fine asbestos. Circular No. 27, 1890, of the Boston Manufacturers' Mutual Fire Insurance Company gives some interesting data in this connection. A moderate layer of soot over the heating surfaces of a boiler or superheater cancels in large measure any gain made by skilful firing or excellence in the design of the engine. Mr. Jaques Abady, of Alex. Wright & Co., London, in a lecture before the Nottingham Guild of Mechanical and Electrical Engineers, rates the conductivity of steel plates covered with a deposit of soot one-sixteenth inch thick as but seventy-five per cent of that of a clean tube. In three tests made by Mr. J. J. Coughlin, Engineer of the Champion Coated Paper Co., Hamilton, Ohio, and reported in *Power* for July 11, 1911, the gain from the use of a soot cleaner is shown to range from 3 to 10.6 per cent under varying conditions.

So great is the loss in this regard that permanent soot blowers are at present attached to most boilers and superheaters. The frequent use of such apparatus is necessary in order to maintain the maximum efficiency of steam generators. Holes through the brickwork, hollow stay bolts and piping inside the setting provide the necessary means for introducing steam or air jets. Soot and ash doors are inserted where needed for removing the residue in the space behind the bridge wall. With certain coals the soot deposited in fire tubes forms a tough scale which resists the action of the steam jet. A cleaning implement consisting of a scraper and brush combined must then be forced through the tubes by hand.

9. Type of Boiler. — A great variety of reasons suggest the adoption of one type of boiler in preference to another. It is not possible to make an absolute comparison of the relative advantages of boilers in general nor is it wise to consider the cost of a boiler, either bare or installed for duty, a final criterion of its superiority. Too many factors enter into the generation of steam to permit so easy an answer to a perplexing question. The present and future conditions of the plant should be as fully considered as possible in seeking a type of boiler for the most successful operation.

The first cost of the horizontal return tubular boiler, exclusive of its setting, is probably the lowest of that of any type. It can be rapidly built and promptly shipped to its destination. Since it contains no separate parts with the exception of the grates and front, it can be transported to its final resting place largely as a unit. Skilled labor, other than that of an ordinary mason and pipe fitter, is not required in its erection or installation. Its setting is neither complicated nor especially liable to deterioration. Relative to its horse-power it occupies a considerable amount of floor space. It has no mud drum, hence the deposits within fall upon the fire sheet, and consequently forbid the use of very bad feed water. The circulation is good and there is ample water surface for the disengagement of dry steam.

It contains a relatively large volume of water which proves likewise a powerful source of energy for sudden calls as well as a tremendous reservoir of destruction in case of accident. It requires but little skill and attention in firing and when fairly handled will endure from twenty to twenty-five years of service with but few repairs. When the latter are necessary, they can frequently be made by ordinary mechanics. It is not as adaptable as the water-tube type for carrying excessive overloads nor has it flexibility enough to commend it for widely varying conditions. The evaporative efficiency is fairly high but this type is generally chosen from other considerations than those of strict economy.

Many of the above statements may be made as well of the Scotch boiler. As a self-contained unit it is easily handled and installed. While its first cost is much more than that of a horizontal return tubular boiler, its setting, consisting merely of two cradles and a covering of magnesia blocks, is correspondingly inexpensive. The circulation is poor and unless special ducts are provided for the purpose, the volume of water below the furnace tube has but little steam-generating capacity. The fire hazard from its internal furnace is small and its shell, removed from intense heat and open to inspection and repair, is rarely known to fail. The furnace tubes are the chief source of difficulty, not providing sufficient grate room and frequently suffering collapse. Firing of the long narrow grate is difficult and refuse coal can rarely be used. Repairs to this type of boiler are expensive and fairly frequent especially if an internal combustion chamber is used. The economy of such boilers is very high.

Vertical boilers as a class have not enough length of circuit from grate to flue. When used in connection with an economizer their efficiency is improved. While easily installed and very economical of room they can rarely be used with poor feed water. The lower tube sheet is a very sensitive member and, even if accessible from adjacent handholes, its cleaning necessitates frequent withdrawals from service. When the outer shell forms a vertical cylinder of uniform diameter throughout, there is room enough outside of the space occupied by the tubes for entrance and inspection. With the shell reduced in diameter at the top of the furnace, the degree of superheating is increased but at the expense of the room for internal inspection. Even though the furnace has sloping sides the circulation in the water leg is poor and the overheating of the neighboring furnace sheet is of frequent occurrence. The wasting of the tubes in the superheating space also makes frequent renewals necessary. As a generator of mildly superheated steam in very close quarters it finds its chief usefulness.

Locomotive boilers of the stationary type are expensive to construct and open to the criticism of inadequate grate area. Unless the rate of combustion is increased either by an exhaust nozzle placed in the stack or by ordinary forced draft, it is difficult to obtain the rated horse-power. The tube ends near the fire suffer from overheating even with baffled walls in use. The renewal of tubes, sheets and stays is attended with considerable expense. Except in cases where their portability on skids makes them valuable, boilers of this type are not widely used. As generators of motive power for railroads, the above disadvantages are far outweighed by the possibilities of power concentration and forcing which locomotive boilers possess. When tubed with flues of small diameter and equipped with superheaters they are fairly economical.

Water-tube boilers, of which there are many types, constitute a class by themselves. They may be considered as steam generators of high cost, great flexibility and strict economy. Since it is rarely possible to ship them already assembled, their erection is a matter of considerable time and expense. They necessitate careful attention and skilful firing. Containing but little water, they suffer rapid fluctuations of pressure with careless handling. With waterways and drums of limited capacity, the rupture of one member is rarely attended by the total destruction of the boiler. They have but little thermal resource in meeting very sudden demands. However, they respond much more rapidly than do the other types to conditions requiring forced working. They can be warmed up and brought into service without injury to their structure in about one-fifth the time required by Scotch boilers. The many joints and cover plates in water-tube boilers require great care in their adjustment and maintenance. Repairs constitute a task demanding special skill. The water circulation is definite and rapid. With the sediment drums usually provided, water of poor quality can be used. As was noted in a previous paragraph boilers of this type permit great power concentration and are capable of carrying severe overloads. Their settings are complicated and unless kept in good repair are liable to permit large losses of economy by air leakage.

10. Circulation. — The necessity for circulation in steam-generating vessels is evident from three standpoints. Water absorbs heat very slowly by conduction. Porcupine boilers with numerous "dead ends" are notoriously inefficient. Convection or current is necessary in the liquid to insure a rapid degree of heat absorption. A very interesting series of experiments was made at the University of Illinois, in 1910, by Messrs. Clement and Garland upon the variation of heat absorption in relation to the velocity of flow of water through steel tubes. The results of these tests are published in Bulletin No. 40 of the Engineering Experiment Station. They show that in general the B.T.U. per minute absorbed by the circulating water per square foot of tube surface is approximately doubled when the circulation increases to eight times its original velocity. While these tests were not performed under conditions entirely similar to those of a water-tube boiler, they nevertheless indicate the value of free circulation in assisting the absorption of heat.

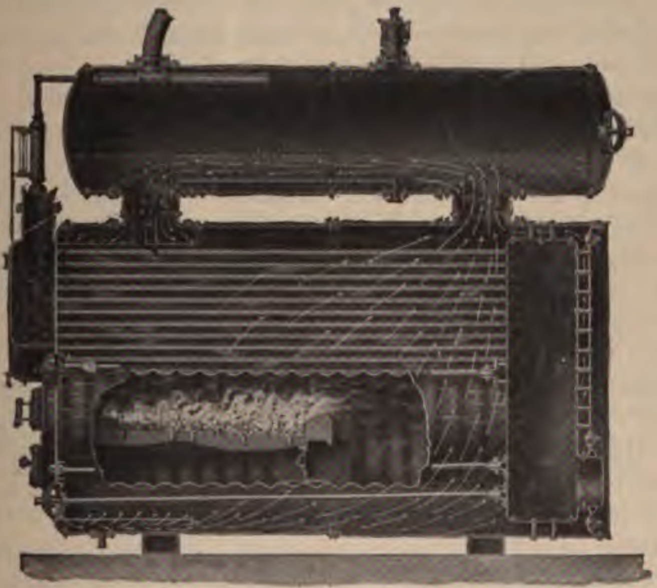
The second necessity for circulation is related to the disengagement of steam. As soon as formed the steam bubbles should be swept from the hot surface and permitted to disengage themselves from the liquid into the steam space without disruptive ebullition. When steam bubbles "pocket" or adhere to the hot plates where they are formed, the heat of the fire is not promptly transmitted, and overheating and burning of the metal wall results. This condition is especially liable to be found in connection with vertical tubes and furnace sheets. Disruptive discharge of steam produces priming and necessitates the use of separators in pipe lines.

The third reason for securing good circulation depends upon the purity of the water. Sediment to be harmless must be kept in motion. If allowed to settle and cake upon the hot sheets, it forms a scale destructive alike to the efficiency of the boiler and the durability of its structure. From the above three standpoints it is evident that definite paths of circulation must be provided and the direction and rapidity of the waterflow insured.

In designing a tube sheet for definite water circulation advantage must be taken of the fact that the hotter the locality, the more numerous will be the steam bubbles and the greater the tendency for them to rise. Hence over the hottest part of the fire there should be vertical channels or avenues to which ascending currents will naturally be attracted. To supply the water for displacing the steam bubbles, provision must be made for descending currents in a cooler part of the boiler. These principles are carried out to a greater or less degree in all well-designed boilers. When horizontal fire tubes are arranged inside of cylindrical shells there is often provided a central circulation space to attract a concourse of the steam bubbles. Especially is this necessary over furnace tubes where ebullition is rapid. Strong descending currents in contact with the cooler shell supply the water necessary to continue the circulation. If the channel through which the circulation takes place is too narrow the progress of the water is arrested and the disengagement of steam is accomplished with violence. On the other hand if the circulation spaces are too wide, local ascending and descending currents are set up which interfere with one another. A width of from three to four inches has been found by experience to constitute a reasonable space where the water velocity is not rapid. Over furnace tubes, spaces from six to eight inches in width are required. The horizontal spaces between the surfaces of ordinary boiler tubes are generally made from one to one and one-quarter inches in width.

When any portion of the water in a boiler fails to participate in the general circulation, not only does that part cease to serve as efficient steam-generating space, but its presence, moreover, induces local stresses in the structure of the boiler very great in amount. Such is frequently the case with the space below the furnace tubes in Scotch boilers. The water in this locality is in contact with the ash-pan sheets above and with the cool outer shell below. It is, therefore, more or less isolated in its position. The stresses set up in the furnace tubes are very severe, due to the fact that the upper half is in contact with the hottest part of the fire and the lower half with a body of comparatively cool water.

The property which certain waters possess of precipitating sediment at temperatures below boiling may be turned to good account in arranging the circulation. Mud drums may be so located as to entrap the deposit, its weight and centrifugal force tending to separate it from the current of water. In water-tube boilers for instance the incoming feed is used to assist the circulation in a downward direction through the rear headers. A mud drum is situated at the lowest angle of the tubes where the motion of the water suddenly changes its direction. This is the logical location for such sediment catchers. A large percentage of the feed-water impurities are thus thrown out of the circulation.



One of the best examples of the use of special apparatus in constraining the circulation to follow definite paths is found in certain types of Scotch boilers. The accompanying half tone illustrates a form of boiler manufactured by the International Engineering Works, South Framingham, Mass. A horizontal drum of comparatively small diameter is joined to the top of the main boiler by flanged openings at the front and rear. The lower shell contains two corrugated furnace tubes, numerous smoke tubes and a commodious combustion chamber. The normal water level is maintained a little below the center of the upper drum. In this manner abundant steam and water volumes are secured. To insure circulation underneath the furnace tubes an annular conduit is riveted to the inside of the lower shell near its front end. All the descending currents from the upper drum pass through this conduit and are liberated at the very lowest point in the boiler between the furnace tubes. In this manner all the water returning from the upper drum is constrained to pass through the space below the furnace tubes. A detailed description of this apparatus is published in Bulletin No. 7, issued by the above concern.

A test made upon one of these boilers at the Sewerage Pumping Plant, South Framingham, Mass., affords some interesting data in regard to temperatures. Four thermometers, installed in as widely separated portions of the shell as possible, were used to indicate the effectiveness of the circulating appliance. The specific location of the thermometers was as follows: No. 1 just behind the rear connecting flange; No. 2 just behind the front connecting flange; No. 3 in the front head below the furnace tubes; and No. 4 in the rear head below the combustion chamber. Observations were taken every five minutes for a period of several hours and the results plotted. Starting with a cool boiler the water above the furnaces and tubes quickly reached the boiling point. As soon as steam began to be liberated the circulation commenced and the temperature beneath the furnace tubes assumed a figure but eight or ten degrees Fahrenheit below that of the hottest portion of the boiler. From this time on all four thermometers were within a very few degrees of one another. It was thus demonstrated that the conduits were effective guides to the circulating currents and caused the water in the boiler to be heated as a whole. Simpler forms of circulation apparatus such as deflecting and baffle plates have been employed with good effect in many types of steam generators.

While the effect of scale and incrustation upon the efficiency of a boiler is a matter of some dispute, there is no question of its effect upon the strength of the tubes and plate. Professor E. B. Schmidt, in Bulletin No. 7 of the University of Illinois, Engineering Experiment Station, concludes from a series of tests made by him upon the heat transference through boiler tubes coated with incrustation ranging up to one-eighth inch in thickness, that the loss in efficiency varies widely, reaching as a maximum ten or twelve per cent. The temperature of a boiler furnace often approaches 2500° F. The safety of the metal in contact with the fire depends upon a transmission of heat so prompt as to keep the plates at a temperature but little in excess of that of the water in the boiler. With deposits of the thickness mentioned above, it is impossible for the circulation to reach the metal and thereby keep it cool. A variety of ills arise from this source, consisting of burnt and wasted plates, sagging sheets and leaky joints. The most available preventative for scale formation, aside from pure feed water, is rapid circulation, combined with a prompt blowing out of the sediment as soon as it settles. For this reason as well, therefore, the designer must exercise great care in securing good circulation within the boiler.

11. Materials used in Boiler Design. — Nothing is more important in the design of a successful steam generator than the skilful selection of the materials to be used. Low-pressure apparatus for steam and hot-water heating is for the most part made of cast iron, the different units or sections being connected by tapering thimbles nicely fitted and forced to place.

The temperature of steam at 250 pounds per square inch gage pressure is about 400° F. Adding to this a possible superheat of 200° F., the temperature of high-pressure superheated steam would reach the region of 600° F. The effect of such high temperatures upon the materials used in boiler construction has been the subject of much controversy and experiment. The strength of wrought iron and soft steel is not seriously affected by temperatures below 900° F., although there is a marked diminution in the per cent contraction of area at rupture. Cast iron, while not influenced to any great degree by the temperatures mentioned above, is not a suitable metal for boiler parts. Its low tensile strength, exceeding brittleness and granular action under the operation of calking have caused the American Boiler Manufacturers' Association and others to exclude it long since from use wherever tensile stresses exist. For special parts such as pipe flanges and the headers of water-tube boilers it is still used, when not in contact with superheated steam, for pressures below 160 pounds per square inch. On account of its liability to corrosion, most boiler rules prohibit the use of cast iron for the discs or seats of valves.

The high pressures and temperatures used in power generation call for a material the chief property of which shall be ductility. While strength is a desirable attribute in the structure of any boiler, the varying conditions of contraction and expansion, of shock and overpressure, and of corrosion and overheating demand a material the toughness of which will endure rough usage and give abundant warning before rupture. The high cost of puddled wrought iron together with its comparatively low tensile strength, have served to eliminate it largely from use in boiler shells. At the same time the modern advance in steel making, especially as regards toughness and homogeneity, has led to the wide adoption of this metal as the material best fitted for use in pressure vessels. With the accuracy now obtainable in determining and placing the carbon content of soft steel, the latter material may be forged and welded with all the reliability possessed by the best wrought iron. With the recent improvements in pressing and extruding metals, it may be fairly stated, that at present every part and portion of a steam boiler may be shaped from soft steel without adding inordinately to the expense. When the cost must be kept low, the modern methods of the steel foundry and annealing oven render the steel casting almost as homogeneous and ductile as the forging. Most boilers, therefore, for severe service and high pressure, are made throughout of soft steel. The larger and more important parts are pressed to shape, either hot or cold, while the less important ones are made of well annealed steel castings.

A copy of the requirements of the American Society for Testing Materials in relation to the steels used in the manufacture of boilers is appended at the end of this article.

At one time copper was widely used for the furnace sheets of locomotive and vertical boilers on account of its ductility and immunity from corrosion. In the form of rolled plates, the tensile strength of copper is 30,000 pounds per square inch, its elastic limit one-half this figure, and the ultimate extension from 40 to 50 per cent. Steel has largely supplanted copper for this purpose, however, because of its relative cheapness and ability to resist high temperatures. Beyond 500° F. the tensile strength of copper rapidly diminishes. The thickness of copper plates required to hold modern high pressures would, therefore, be prohibitive. Thin copper ferrules are widely used to make the joint between the tubes and tube sheets of locomotive boilers. The non-corrosive property of the copper permits a little flexibility of the joint while preventing leakage.

For general purposes in boiler-feed piping and accessories various brass alloys are employed. Where the feed pipe enters the boiler a brass bushing should be used to withstand, as far as possible, the vibration of the feed pump as well as to prevent corrosion. United States Navy composition, consisting of copper 88 per cent, tin 10 per cent, and zinc 2 per cent, makes a good but expensive alloy for this purpose. Its tensile strength is about 32,000 pounds per square inch and its ultimate elongation 25 per cent. To resist the corrosive action of sea water in marine condensers and fittings, many alloys have been used. Tobin bronze in the form of plates and rivets may be readily forged at a red heat. Its component parts are copper 60 per cent, zinc 38 per cent, tin 1.5 per cent, iron 0.17 per cent, lead 0.33 per cent. The tensile strength of Tobin bronze in the form of hot rolled rods is about 80,000 pounds per square inch and the elastic limit 50,000 pounds per square inch. In the form of plates the tensile strength is taken as 50,000 pounds per square inch and the elastic limit 30,000 pounds per square inch.

For the cores of fusible plugs the United States Government Specifications call for the use of pure Banca Tin, the fusing point of which is about 445° F. This temperature is well in advance of that of saturated steam at the highest practical pressure. The use of lead for gaskets in the region of the fire is not to be recommended since its low fusing point, 625° F., renders it unsafe for such a purpose.

STANDARD SPECIFICATIONS FOR BOILER AND FIREBOX STEEL.

Adopted August 25, 1913.
Revised September 10, 1914.

1. These specifications cover two grades of steel for boilers, namely: flange and firebox.

I. MANUFACTURE.

2. The steel shall be made by the open-hearth process.

II. CHEMICAL PROPERTIES AND TESTS.

3. The steel shall conform to the following requirements as to chemical composition:

		Flange.	Firebox.
Carbon, per cent		....	0.12–0.25
Manganese, per cent		0.30–0.60	0.30–0.50
Phosphorus	Acid, per cent	not over 0.05	not over 0.04
	Basic, per cent	" " 0.04	" " 0.035
Sulphur, per cent		" " 0.05	" " 0.04
Copper, per cent		....	" " 0.05

4. An analysis shall be made by the manufacturer from a test ingot taken during the pouring of each melt, a copy of which shall be given to the purchaser or his representative. This analysis shall conform to the requirements specified in Section 3.

5. Analyses may be made by the purchasers from a broken tension test specimen representing each plate as rolled, which shall conform to the requirements specified in Section 3.

III. Physical Properties and Tests.

6. (*a*) The material shall conform to the following requirements as to tensile properties:

	Flange.	Firebox.
Tensile strength, lbs. per sq. in.	55,000–65,000	52,000–62,000
Yield point, min. lbs. per sq. in.	0.5 tens. str.	0.5 tens. str.
Elongation in 8 in., min. per cent (See Section 7)	$\frac{1,500,000}{\text{Tens. str.}}$	$\frac{1,500,000}{\text{Tens. str.}}$

(*b*) The yield point shall be determined by the drop of the beam of the testing machine.

7. (*a*) For material over ¾ in. in thickness, a deduction of 0.5 from the percentages of elongation specified in Section 6 (*a*) shall be made for each increase of ⅛ in. in thickness above ¾ in.

(*b*) For material ¼ in. or under in thickness, the elongation shall be measured on a gage length of 24 times the thickness of the specimen.

8. (*a*) *Cold-bend Tests.* — The test specimen shall bend cold through 180 degrees without cracking on the outside of the bent portion, as follows: For material 1 in. or under in thickness, flat on itself; and for material over 1 in. in thickness, around a pin the diameter of which is equal to the thickness of the specimen.

(*b*) *Quench-bend Tests.* — The test specimen, when heated to a light cherry red as seen in the dark (not less than 1200° F.) and quenched at once in water, the temperature of which is between 80° and 90° F., shall bend through 180 degrees without cracking on the outside of the bent portion, as follows: For material 1 in. or under in thickness, flat on itself; and for material over 1 in. in thickness, around a pin the diameter of which is equal to the thickness of the specimen.

9. For firebox steel, a sample taken from a broken tension test specimen shall not show any single seam or cavity more than ¼ in. long, in either of the

three fractures obtained in the test for homogeneity, which shall be made as follows:

The specimen shall be either nicked with a chisel or grooved on a machine, transversely, about $\frac{1}{16}$ in. deep, in three places about 2 ins. apart. The first groove shall be made 2 ins. from the square end; each succeeding groove shall be made on the opposite side from the preceding one. The specimen shall then be firmly held in a vise, with the first groove about $\frac{1}{4}$ in. above the jaws, and the projecting end broken off by light blows of a hammer, the bending being away from the groove. The specimen shall be broken at the other two grooves in the same manner. The object of this test is to open and render visible to the eye any seams due to failure to weld up or to interposed foreign matter, or any cavities due to gas bubbles in the ingot. One side of each fracture shall be examined and the lengths of the seams and cavities determined, a pocket lens being used if necessary.

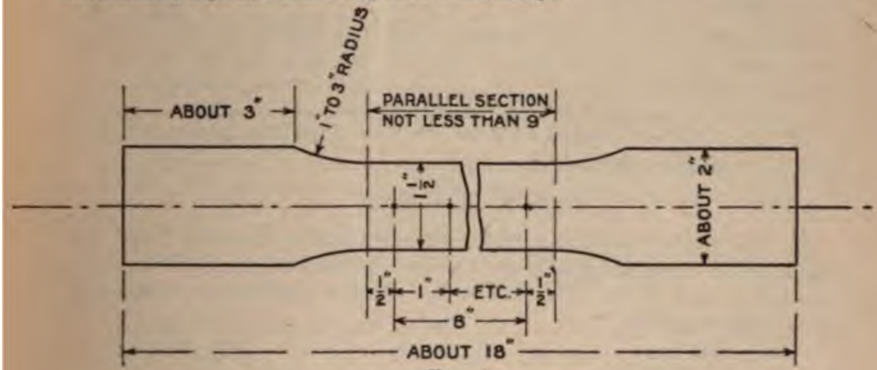


FIG. 1.

10. Tension and bend test specimens shall be taken from the finished rolled material. They shall be of the full thickness of material as rolled, and shall be machined to the form and dimensions shown in Fig. 1; except that bend test specimens may be machined with both edges parallel.

11. (*a*) One tension, one cold bend, and one quench-bend test shall be made from each plate as rolled.

(*b*) If any test specimen shows defective machining or develops flaws, it may be discarded and another specimen substituted.

(*c*) If the percentage of elongation of any tension test specimen is less than that specified in Section 6 (*a*) and any part of the fracture is outside the middle third of the gage length, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed.

IV. PERMISSIBLE VARIATIONS IN WEIGHT AND GAGE.

12. *When Ordered to Gage.* — The thickness of each plate shall not vary more than 0.01 in. under that ordered.

An excess over the nominal weight corresponding to the dimensions on the order shall be allowed for each plate, if not more than that shown in the following table, one cubic inch of rolled steel being assumed to weigh 0.2833 lb.:

Thickness ordered. Ins.	Nominal weight. Lbs. per sq. ft.	Allowable excess (expressed as percentage of nominal weight).						
		Under 50 ins.	50 to 70 ins. excl.	70 ins. or over.	Under 75 ins.	75 to 100 ins. excl.	100 to 115 ins. excl.	115 ins. or over.
$\frac{1}{8}$ to $\frac{5}{32}$	5.10 to 6.37	10	15	20	....	....	....	....
$\frac{5}{32}$ to $\frac{3}{16}$	6.37 to 7.65	8.5	12.5	17	....	....	....	....
$\frac{3}{16}$ to $\frac{1}{4}$	7.65 to 10.20	7	10	15	....	....	....	....
$\frac{1}{4}$	10.20	....	....	....	10	14	18	....
$\frac{5}{16}$	12.75	....	....	....	8	12	16	....
$\frac{3}{8}$	15.30	....	....	....	7	10	13	17
$\frac{7}{16}$	17.85	....	....	....	6	8	10	13
$\frac{1}{2}$	20.40	....	....	....	5	7	9	12
$\frac{9}{16}$	22.95	....	....	....	4.5	6.5	8.5	11
$\frac{5}{8}$	25.50	....	....	....	4	6	8	10
Over $\frac{5}{8}$	....	....	....	....	3.5	5	6.5	9

V. FINISH.

13. The finished material shall be free from injurious defects and shall have a workmanlike finish.

VI. MARKING.

14. The name or brand of the manufacturer, melt or slab number, grade, and lowest tensile strength for its grade specified in Section 6 (*a*), shall be legibly stamped on each plate. The melt or slab number shall be legibly stamped on each test specimen.

VII. INSPECTION AND REJECTION.

15. The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests (except check analyses) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

16. (*a*) Unless otherwise specified, any rejection based on tests made in accordance with Section 5 shall be reported within five working days from the receipt of samples.

(*b*) Material which shows injurious defects subsequent to its acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

17. Samples tested in accordance with Section 5, which represents rejected material, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

STANDARD SPECIFICATIONS FOR BOILER RIVET STEEL.

Adopted August 25, 1913.
Revised September 10, 1914.

A. REQUIREMENTS FOR ROLLED BARS.

I. MANUFACTURE.

1. The steel shall be made by the open-hearth process.

II. CHEMICAL PROPERTIES AND TESTS.

2. The steel shall conform to the following requirements as to chemical composition:

Manganese, per cent	0.30 to 0.50
Phosphorus, " "	not over 0.04
Sulphur, " "	" " 0.045

3. An analysis to determine the percentages of carbon, manganese, phosphorus and sulphur shall be made by the manufacturer from a test ingot taken during the pouring of each melt, a copy of which shall be given to the purchaser or his representative. This analysis shall conform to the requirements specified in Section 2.

4. Analyses may be made by the purchaser from finished bars representing each melt, which shall conform to the requirements specified in Section 2.

III. PHYSICAL PROPERTIES AND TESTS.

5. (*a*) The bars shall conform to the following requirements as to tensile properties:

Tensile strength, lbs. per sq. in.	45,000 to 55,000
Yield point, min. lbs. per sq. in.	0.5 tens. str.
Elongation in 8 in., min. per cent	$\frac{1{,}500{,}000}{\text{Tens. str.}}$

but need not exceed 30 per cent.

(*b*) The yield point shall be determined by the drop of the beam of the testing machine.

6. (*a*) *Cold-bend Tests.* — The test specimen shall bend cold through 180 degrees flat on itself without cracking on the outside of the bent portion.

(*b*) *Quench-bend Tests.* — The test specimen, when heated to a light cherry red as seen in the dark (not less than 1200° F.), and quenched at once in water the temperature of which is between 80° and 90° F., shall bend through 180 degrees flat on itself without cracking on the outside of the bent portion.

7. Tension and bend test specimens shall be of the full-size section of bars as rolled.

8. (*a*) Two tension, two cold-bend, and two quench-bend tests shall be made from each melt, each of which shall conform to the requirements specified.

(*b*) If any test specimen develops flaws, it may be discarded and another specimen substituted.

(*c*) If the percentage of elongation of any tension test specimen is less than that specified in Section 6 (*a*) and any part of the fracture is outside the middle third of the gage length, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed.

IV. PERMISSIBLE VARIATIONS IN GAGE.

9. The gage of each bar shall not vary more than 0.01 in. from that specified.

V. WORKMANSHIP AND FINISH.

10. The finished bars shall be circular within 0.01 in.

11. The finished bars shall be free from injurious defects and shall have a workmanlike finish.

VI. MARKING.

12. Rivet bars shall, when loaded for shipment, be properly separated and marked with the name or brand of the manufacturer and the melt number for identification. The melt number shall be legibly marked on each test specimen.

VII. INSPECTION AND REJECTION.

13. The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the bars ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the bars are being furnished in accordance with these specifications. All tests (except check analyses) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

14. (*a*) Unless otherwise specified, any rejection based on tests made in accordance with Section 4 shall be reported within five working days from the receipt of samples.

(*b*) Bars which show injurious defects subsequent to their acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

15. Samples tested in accordance with Section 4, which represent rejected bars, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

B. REQUIREMENTS FOR RIVETS.

I. PHYSICAL PROPERTIES AND TESTS.

16. The rivets when tested shall conform to the requirements as to tensile properties specified in Section 5, except that the elongation shall be measured on a gage length not less than four times the diameter of the rivet.

17. The rivet shank shall bend cold through 180 degrees flat on itself, as shown in Fig. 2, without cracking on the outside of the bent portion.

18. The rivet head shall flatten, while hot, to a diameter 2½ times the diameter of the shank, as shown in Fig. 3, without cracking at the edges.

19. (*a*) When specified, one tension test shall be made from each size in each lot of rivets offered for inspection.

(*b*) Three bend and three flattening tests shall be made from each size in each lot of rivets offered for inspection, each of which shall conform to the requirements specified.

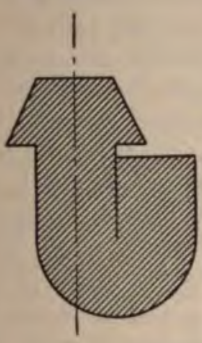


Fig. 2.

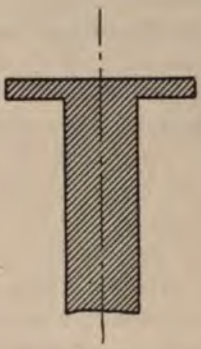


Fig. 3.

II. Workmanship and Finish.

20. The rivets shall be true to form, concentric, and shall be made in a workmanlike manner.

21. The finished rivets shall be free from injurious defects.

III. Inspection and Rejection.

22. The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the rivets ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the rivets are being furnished in accordance with these specifications. All tests and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

23. Rivets which show injurious defects subsequent to their acceptance at the manufacturer's works will be rejected and the manufacturer shall be notified.

12. Factor of Safety. — Many of the conditions which must be considered in the selection of a factor of safety are mentioned in the foregoing paragraphs. Liability to shock from water hammer, lack of homogeneity in the material, deterioration due to corrosion, errors of workmanship, the interdependence of parts, the hazard to human life, and the probability of severe overload summarize the principal reasons for care in choosing the margin of safety for pressure vessels. To these may be added stresses due to the method of suspension, especially in the case of locomotive boilers.

There are two general methods of specifying the value for the factor of safety. First, a variety of factors may be stipulated, some depending upon the length of time the boiler has been in commission and others taking into account the peculiarities of certain types of design. To these should be added one general value for new work. This is the method adopted in general by the Federal and State Governments of the United States. The second method consists of specifying one base value for all classes of work to which shall be added certain amounts determined by age, faulty construction, poor design, and various stages of repair. These increments in the factor of safety are cumulative and it is the plan to require so great a value for poor and dangerous constructions as to discourage their use. The Dominion of Canada pursues the latter method.

The Massachusetts Boiler Rules have been used as a model by many states and adopted bodily by others. Their stipulations in regard to factor of safety are as follows:

For new boilers	5.00
For stays and stay bolts (new work)	6.50

For boilers in the state at the time of the enactment of the rules but not passing Massachusetts inspection, if used thereafter, minimum factors of safety are to be as follows:

Boilers with longitudinal lap joints not exposed to combustion	6.00
Boilers with longitudinal lap joints not over 36 inches in diameter but exposed to combustion	6.00
Boilers with longitudinal lap joints, over 36 inches in diameter and exposed to combustion	8.00
Boilers with butt-strap joints not over 10 years old	4.50
Boilers with butt-strap joints over 10 years old	5.00

The test pressure is in all cases to be one and one-half times the working pressure.

The base value for the Canadian rule is 4.50 for boilers of approved design. To this is added a series of increments ranging from 1.00 for the use of single-riveted lap joints to 0.70 for double-riveted lap joints and 0.50 for triple-riveted lap joints. Various other penalties are exacted ranging down to 0.07 for poor alignment of holes. The total factor with the worst possible accumulation of bad practice reaches 8.50, which compares favorably with that of the Massachusetts standard for longitudinal lap joints when used upon externally fired boilers over 36 inches in diameter.

United States Federal rules do not specify an exact value for the factor of safety but prescribe an elaborate set of limits for use in connection with various materials and forms. These rules are contained in the manual published by the United States Supervising Inspectors of Steam Vessels.

13. Massachusetts Boiler Rules. — The following is a brief summary of those portions of the Massachusetts Rules which directly affect the design of steam boilers.

1. Boilers exempt from inspection:
   - (*a*) Boilers of railroad locomotives.
   - (*b*) Boilers of motor road vehicles.
   - (*c*) Boilers in private residences.
   - (*d*) Boilers under jurisdiction of the United States.
   - (*e*) Boilers used for agricultural purposes only.
   - (*f*) Boilers of less than three horse-power.
   - (*g*) Boilers used for heating purposes solely when equipped with a district police lock pop safety valve set at 15 pounds or less, and having less than 4 square feet of grate surface.
   - (*h*) Fire engine boilers temporarily in the state during conflagrations.

2. All other boilers shall be inspected when installed and annually thereafter by the Inspection Department of the District Police under supervision of the Chief Inspector of Boilers.

3. Boilers shall be made ready for examination upon fourteen days notice unless adjudged to be in a dangerous condition, in which case immediate discontinuance from service may be required.

4. No insurance shall be effective upon any boiler installed after the enactment of these laws unless the boiler conforms to said standards.

5. All boilers shall be equipped with one or more fusible safety plugs located as specified hereafter.

6. All boilers of special design shall be subjected directly to the Board for approval.

The Massachusetts Boiler Rules are divided into three parts as follows:

PART I. Rules applicable to boilers installed before the enactment of these laws.

PART II. Rules applicable to all boilers whenever installed.

PART III. Rules applicable to boilers installed after the enactment of these laws.

PART I.

Applicable to boilers installed before the enactment of these laws.

7. The allowable working pressure for a shell or drum boiler shall be determined by reference to the tensile strength and thickness of the plate,

the efficiency of the longitudinal joints, and the diameter of the shell, the units being pounds and inches throughout:

$$\text{Pressure} = \frac{\text{Tens. str. of plate} \times \text{Thickness} \times \text{Eff. of joint}}{\text{Inside radius of outer course} \times \text{Factor of safety}}.$$

8. When unknown, the tensile strength shall be assumed to be 55,000 pounds per square inch for steel plate and 45,000 pounds per square inch for wrought iron.

9. Factor of safety for boilers with double-covered butt joints shall be assumed to be four and five-tenths (4.5).

When unknown, the diameter of rivets shall be assumed as follows:

Plate thickness. Ins.	Assumed driven diam. of rivets. Ins.
1/4	11/16
9/32	11/16
5/16	3/4
11/32	3/4
3/8	3/4 up to and including 2 ins. pitch.
3/8	13/16 over 2 ins. pitch.
13/32	13/16
7/16	7/8 up to and including 2¼ ins. pitch.
7/16	15/16 over 2¼ ins. pitch.
15/32	15/16
1/2	15/16
9/16	1 1/16
5/8	1 1/16

10. The size of safety valves, other than spring loaded, shall be governed by the pressure and grate area according to the following table:

Maximum pressure allowed per square inch on the boiler.	Zero to 25 pounds, incl.	Over 25 to 50 pounds, incl.	Over 50 to 100 pounds, incl.
Diameter of Valve. Inches.	Area of grate. Square feet.		
1	1.50	1.75	2.00
1¼	2.25	2.50	3.00
1½	3.00	3.75	4.00
2	5.50	6.50	7.25
2½	8.25	10.00	11.00
3	11.75	14.25	16.00
3½	16.00	19.50	21.75
4	21.00	25.50	28.25
4½	26.75	32.50	36.00
5	32.75	40.00	44.00

11. The conditions of installment for safety valves upon boilers in battery, connected to a common steam main, shall be such that low-pressure boilers can never be accidentally loaded with steam at a pressure beyond the allowable amount.

12. Each boiler shall have a blow-off pipe connected with the lowest available water space.

PART II.

Applicable to all boilers whenever installed.

13. The pressure allowed on a boiler constructed wholly of cast iron shall not exceed 25 pounds per square inch.

14. The pressure on water-tube boilers, the tubes of which are secured to cast iron headers, shall not exceed 160 pounds per square inch.

15. The crushing resistance of soft steel plate shall be taken as 95,000 pounds per square inch of projected area.

16. The shearing strength of rivets per square inch of sectional area shall be taken as follows:

	Lbs.
Iron rivets, single shear	38,000
Iron rivets, double shear	70,000
Steel rivets, single shear	42,000
Steel rivets, double shear	78,000

17. The lowest factors of safety used for boilers, the shells or drums of which are exposed to the products of combustion and the longitudinal joints of which are of lap-riveted construction, shall be as follows:

(*a*) Five (5) for boilers not over ten years old.

(*b*) Five and five-tenths (5.5) for boilers over ten and not over fifteen years old.

(*c*) Five and seventy-five hundredths (5.75) for boilers over fifteen and not over twenty years old.

(*d*) Six (6) for boilers over twenty years old.

(*e*) Five (5) for boilers the longitudinal joints of which are of lap-riveted construction and the shells or drums of which are not exposed to the products of combustion.

18. Each boiler shall have one or more safety valves.

19. The size of spring-loaded safety valves shall be determined by reference to the pressure and grate area of the boiler, according to the table on the following page.

20. The conditions of installment for spring-loaded safety valves upon boilers in battery, connected to a common steam main, shall be such that low-pressure boilers can never be accidentally loaded with steam at a pressure beyond the allowable limit.

21. When conditions exceed those in the above table, spring-loaded safety valves shall be calculated by the following formula, the units being as stated:

$$\text{Area of valve, sq. in.} = \frac{770 \times \text{Lbs. water evap. per sq. ft. of grate per sec.}}{\text{Abs. pressure at which valve is to blow, lbs. sq. in.}}$$

Maximum pressure allowed per square inch on the boiler.	Zero to 25 pounds, incl.	Over 25 to 50 pounds, incl.	Over 50 to 100 pounds, incl.	Over 100 to 150 pounds, incl.	Over 150 to 200 pounds, incl.	Over 200 pounds.
Diameter of valve. Inches.	Area of grate. Square feet.					
1	2.00	2.50	2.75	3.25	3.50	3.75
1¼	3.25	4.00	4.25	5.00	5.50	5.75
1½	4.50	5.50	6.00	7.25	8.00	8.50
2	8.00	9.75	10.75	13.00	14.00	15.00
2½	12.50	15.00	16.50	20.00	22.00	23.00
3	17.75	21.50	24.00	29.00	31.50	33.25
3½	24.00	29.50	32.50	39.50	43.00	45.25
4	31.50	38.25	42.50	51.50	56.00	59.00
4½	40.00	48.50	53.50	65.00	71.00	74.25
5	49.00	60.00	66.00	80.00	88.00	92.25

22. The combined areas of several safety valves when used shall conform to the requirements of the above table.

23. Safety valve connections and escape pipes shall be full size throughout and no valve of any kind shall be placed between the safety valve and the boiler nor on the escape pipe between the safety valve and the atmosphere. All escape pipes shall be fitted with an open drain.

24. Safety valves having either seat or disc of cast iron shall not be used.

25. Safety valves hereafter installed on boilers shall not exceed five (5) inches in diameter, shall be spring loaded with seat at about forty-five (45) degrees to the center point of the spindle and shall be designed with a lifting device so that the disc can be lifted from its seat not less than one-eighth (⅛) the diameter of the valve with the pressure at seventy-five (75) per cent of that at which the safety valve is set to blow.

26. Fusible plugs shall be filled with pure tin.

27. The least diameter of fusible metal shall not be less than one-half (½) inch, except for working pressures of over one hundred and seventy-five (175) pounds, or when it is necessary to place a fusible plug in a tube, in which cases the least diameter of fusible metal shall not be less than three-eighths (⅜) inch.

28. Each boiler shall have one or more fusible plugs of length sufficient to project through the sheet not less than one (1) inch and located as follows:

(a) *Horizontal return tubular boilers* — in rear head not less than two (2) inches above upper surface of top row of tubes.

(b) *Locomotive type boilers* — in the highest part of the crown sheet.

(c) *Vertical fire-tube boilers* — in an extra heavy outside tube not less than one-third (⅓) the tube length above lower tube sheet.

(d) *Vertical submerged tube sheet boiler* — in upper tube sheet.

(e) *Water-tube boilers with horizontal drums* — in upper drum not less than six (6) inches above the bottom of the drum over the first pass of the flames.

(f) *Scotch marine boilers* — in combustion chamber top.

(*g*) *Dry-back Scotch boilers* — in rear head not less than two (2) inches above the surface of the upper row of tubes.

(*h*) For other types and designs fusible plugs shall in general be placed at the lowest permissible water level, in the direct path of the flames, as near the primary combustion chamber as possible.

29. Each boiler shall have a steam gage connected by siphon with T-handle shut-off cock, the gage to read to a pressure not less than one and one-half (1½) times that allowed for the boiler.

30. Each boiler shall have at least one (1) water glass the lowest visible part of which shall be above the level of the fusible plug. In addition two (2) or more gage cocks shall be provided for boilers whose maximum pressure does not exceed twenty-five (25) pounds per square inch, the same to be located within the visible range of the water glass. In the case of boilers designed for more than twenty-five (25) pounds per square inch pressure, three (3) gage cocks shall be employed, the location to be similar to those specified above.

31. Each boiler shall have a feed pipe fitted with a check and stop valve, the latter to be placed between the check valve and the boiler, and the feed water to discharge below the lowest safe water line.

32. A boiler having one (1) square foot of grate surface shall be rated at three (3) horse-power when the safety valve is set to blow at over twenty-five (25) pounds pressure per square inch.

33. A boiler having two (2) square feet of grate surface shall be rated at three (3) horse-power when the safety valve is set to blow at twenty-five (25) pounds pressure per square inch, or less.

34. Hydrostatic pressure test shall not exceed one and one-half (1½) times the working pressure except in the case of pipe boilers and boilers permitted to carry not over twenty-five (25) pounds per square inch. In such cases twice the working pressure may be employed.

35. The efficiency of any type of riveted joint shall be determined as follows:

$$\text{Efficiency} = \frac{\text{Least resistance to failure per unit length of joint}}{\text{Strength of solid plate per unit length of joint}}.$$

PART III.

Applicable to boilers installed after the enactment of these laws.

36. The material used for plates and rivets in the construction of steel shells or drums of boilers shall be as specified by the American Society for Testing Materials.

37. Steel castings in boiler parts shall have not less than fifty thousand (50,000) pounds per square inch tensile strength.

38. Cast iron where permitted in boiler parts shall have not less than eighteen thousand (18,000) pounds per square inch tensile strength.

39. Cross pipes, mud and water drums, and all other connections where the pressure exceeds one hundred and sixty (160) pounds per square inch shall be of wrought or cast steel.

40. Pressure parts of superheaters, whether attached or separately fired, shall be of wrought or cast steel when the working pressure exceeds fifty (50) pounds per square inch.

41. Boiler and superheater mountings shall be of wrought or cast steel when exposed to steam which is superheated over eighty (80) degrees Fahrenheit.

42. Waterleg and door-frame rings of vertical boilers thirty-six (36) inches or over in diameter shall be of wrought or cast steel or wrought iron.

43. Waterleg and door-frame rings of locomotive-type boilers shall be of wrought or cast steel or wrought iron.

44. The working pressure of new boilers shall be based on a factor of safety of five (5).

45. The tube sheets of horizontal return tubular boilers not over thirty-six (36) inches in diameter and designed for not over one hundred (100) pounds per square inch pressure may be stayed by steel angles or T-bars riveted to place, the above angles or T-bars to be calculated as uniformly loaded beams with a fiber stress not exceeding sixteen thousand (16,000) pounds per square inch.

46. The longitudinal joints of a boiler the shell or drum of which exceeds thirty-six (36) inches in diameter shall be of butt and double strap construction.

47. The longitudinal joints of a boiler the shell or drum of which does not exceed thirty-six (36) inches in diameter may be of lap-riveted construction, and the working pressure on such shells shall not exceed one hundred (100) pounds per square inch.

48. All longitudinal joints of horizontal return tubular boilers shall be located above the fire line of the setting.

49. Longitudinal joints shall not have a continuous length of over twelve (12) feet. All shell plates shall be of the same gage with a minimum of one-fourth ($\frac{1}{4}$) inch.

50. The minimum thickness of shell plates shall be as follows:

When the diameter of shell is —			
36 inches or under.	Over 36 to 54 inches inclusive.	Over 54 to 72 inches inclusive.	Over 72 inches.
$\frac{1}{4}$ inch	$\frac{5}{16}$ inch	$\frac{3}{8}$ inch	$\frac{1}{2}$ inch

51. The minimum thickness of butt straps shall be determined by the following table, straps to be accurately rolled to form:

Thickness of shell plates. Ins.	Minimum thickness of butt straps. Ins.	Thickness of shell plates. Ins.	Minimum thickness of butt straps. Ins.
$\frac{1}{4}$	$\frac{1}{4}$	$\frac{17}{32}$	$\frac{7}{16}$
$\frac{9}{32}$	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{7}{16}$
$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$
$\frac{11}{32}$	$\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$
$\frac{3}{8}$	$\frac{5}{16}$	$\frac{7}{8}$	$\frac{5}{8}$
$\frac{13}{32}$	$\frac{5}{16}$	1	$\frac{3}{4}$
$\frac{7}{16}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{3}{4}$
$\frac{15}{32}$	$\frac{3}{8}$	$1\frac{1}{4}$	$\frac{7}{8}$
$\frac{1}{2}$	$\frac{7}{16}$		

52. The minimum thickness of tube sheets shall be as follows:

When the diameter of tube sheet is —			
42 inches or under.	Over 42 to 54 inches inclusive.	Over 54 to 72 inches inclusive.	Over 72 inches.
$\frac{3}{8}$ inch	$\frac{7}{16}$ inch	$\frac{1}{2}$ inch	$\frac{9}{16}$ inch

53. The minimum thickness of a convex head shall be determined as follows, all units being in inches and pounds:

$$\text{Thickness} = \frac{\text{Half radius of convexity} \times \text{Factor of safety} \times \text{Work. pressure}}{\text{Tens. str. of material}}.$$

54. The thickness of concave heads shall be one and two-thirds ($1\frac{2}{3}$) times that of the corresponding convex heads and a factor of safety of five (5) shall be used in such calculations.

55. Convex or concave heads with a manhole opening shall have a thickness at least one-eighth ($\frac{1}{8}$) of an inch greater than that calculated above.

56. Convex or concave heads with a manhole opening shall have the flange turned inward to a depth of not less than three (3) times the thickness of the head.

57. The spacing of stay bolts with riveted heads for use with flat plates or cylindrical furnace sheets of internally fired boilers in which the external diameter of the furnace is over thirty-eight (38) inches shall conform to that given in the following table:

TABLE OF MAXIMUM ALLOWABLE PITCH OF SCREWED STAY BOLTS, ENDS RIVETED OVER.

Pressure, in pounds per square inch.	Thickness of plate.						
	5/16 inch	3/8 inch	7/16 inch	1/2 inch	9/16 inch	5/8 inch	11/16 inch
	Maximum pitch of stay bolts, Ins.						
100	5½	6¼	7	7¾	8½	......	......
110	5¼	6	6¾	7⅜	8⅛	......	......
120	5⅛	5¾	6½	7⅛	7⅞	8½	......
125	5	5⅝	6⅜	7	7¾	8⅜	......
130	5	5⅝	6¼	6⅞	7⅝	8¼	......
140	4¾	5½	6	6⅝	7⅜	8	......
150	4¾	5¼	5⅞	6½	7⅛	7¾	8⅜
160	4⅝	5⅛	5¾	6¼	6⅞	7½	8
170	4½	5	5⅝	6⅛	6¾	7¼	7⅞
180	4⅜	4⅞	5½	6	6½	7⅛	7¾
190	4⅜	4⅞	5⅜	5⅞	6⅜	7	7½
200	4¼	4¾	5¼	5¾	6¼	6¾	7⅜
225	4⅛	4½	5	5½	6	6½	7
250	4	4⅜	4¾	5¼	5¾	6¼	6⅝
300	3¾	4⅛	4½	4⅞	5⅜	5¾	6⅛

Corrugated or reinforced furnace tubes are not amenable to the dimensions given in the above table. Stay bolts adjacent to furnace doors or other fittings, where the allowable pitch is not over five and one-half (5½) inches, may have an increased pitch of not over one (1) inch.

58. An internally fired boiler, in which the external diameter of the furnace is thirty-eight (38) inches or less, except a corrugated or reinforced furnace, shall have the furnace sheet supported by at least one row of stay bolts, the circumferential pitch of which shall not exceed that given in the following table and the minimum outside diameter of stay bolts shall be as given in the table of paragraph 59.

CIRCUMFERENTIAL PITCH OF STAY BOLTS.

Thickness of Furnace Sheet.	Pressure, in pounds per square inch.							
	100	110	120	125	130	140	150	175
	Circumferential pitch of stay bolts, in inches, not to exceed							
¼ inch.....	4¾	4⅝	4½	4⅜	4⅜	4¼	4⅛	3⅞
5/16 inch....	5½	5¼	5⅛	5	5	4¾	4¾	4½

59. The longitudinal pitch of stay bolts on the furnace sheet of an internally fired boiler, in which the external diameter of the furnace is thirty-eight (38) inches or less, except a corrugated or reinforced furnace, shall not exceed that given in the following table:

LONGITUDINAL PITCH OF STAY BOLTS.

External diameter of furnace not exceeding 20 inches.	Pressure, in pounds per square inch.							
	100	110	120	125	130	140	150	175
	Longitudinal pitch of stay bolts, in inches, not to exceed —							
Thickness of furnace sheet ¼ inch.	12⅜	10¼	8⅝	7⅞	7¼	6⅜	5½	4
	Diameter of stay bolts over threads shall not be less than three-fourths (¾) inch.							

External diameter of furnace not exceeding 26 inches.	Pressure, in pounds per square inch.							
	100	110	120	125	130	140	150	175
	Longitudinal pitch of stay bolts, in inches, not to exceed —							
Thickness of furnace sheet ¼ inch.	7¼	6	5⅛	4⅝	4⅜	....	.....	....
	Diameter of stay bolts over threads shall not be less than three-fourths (¾) inch.							
Thickness of furnace sheet 5/16 inch.	15⅞	14¾	12⅜	11⅜	10⅝	9	8	5¾
	Diameter of stay bolts over threads shall not be less than seven-eighths (⅞) inch.							

External diameter of furnace not exceeding 32 inches.	Pressure, in pounds per square inch.					
	100	110	120	125	130	140
	Longitudinal pitch of stay bolts, in inches, not to exceed —					
Thickness of furnace sheet 5/16 inch.	11¾	9¾	8⅛	7⅝	7	6
	Diameter of stay bolts over threads shall not be less than seven-eighths (⅞) inch.					

External diameter of furnace not exceeding 38 inches.	Pressure, in pounds per square inch.				
	100	110	120	125	130
	Longitudinal pitch of stay bolts, in inches, not to exceed —				
Thickness of furnace sheet 5/16 inch.	8⅜	7	5⅞	5⅜	5
	Diameter of stay bolts over threads shall not be less than seven-eighths (⅞) inch.				

60. When the longitudinal joint of the furnace sheet of a fire-tube boiler is of lap-riveted construction a stay bolt in each row shall be located as near the joint as possible.

61. The upper segment of heads of horizontal boilers more than thirty-six (36) inches in diameter shall be stayed by welded or weldless mild steel or wrought iron through stay rods or diagonal braces.

62. Horizontal return tubular boilers having a manhole below the tubes shall have one or more stays on each side of the manhole, the rear ends of which shall be attached to the rear head of the boiler, and the front ends shall pass through the front head and shall be secured with nuts inside and out. The center line of such stays at the front head shall not be below the center line of the manhole.

63. Stay rods shall not exceed three (3) feet in length when screwed through the sheets and riveted over.

64. The maximum allowable stress per square inch of stays and stay bolts shall be as follows, the same to be based upon a factor of safety of at least six and five-tenths (6.5).

STRESSES IN STAYS.

Material and type.	Size up to and including 1¼ ins. diameter or equivalent area.	Size over 1¼ ins. diameter or equivalent area.
	Lbs. sq. ins.	Lbs. sq. ins.
Weldless mild steel head to head or through stays	8000	9000
Weldless mild steel diagonal or crow-foot stays	7500	8000
Weldless wrought iron head to head or through stays	7000	7500
Weldless wrought iron diagonal or crow-foot stays	6500	7000
Welded mild steel or wrought iron stays	6000	6000
Mild steel or wrought iron stay bolts	6500	7000

65. Stay bolts shall be threaded with United States standard threads either ten (10) or twelve (12) to the inch and their strength shall be based upon their root area and the allowable fibre stresses given in the above table.

66. The minimum thickness of cast iron nozzles shall be determined by reference to the working pressure, inside diameter, factor of safety and tensile strength of cast iron, the units being pounds and inches throughout:

$$\text{Thickness} = \frac{\text{Pressure} \times \text{Inside diameter}}{3000} + 0.5.$$

The above expression is based upon the tensile strength of eighteen thousand (18,000) pounds per square inch previously recommended for cast iron and a factor of safety of twelve (12).

67. When the pressure exceeds one hundred and thirty-five (135) pounds per sq. in., the thickness of cast iron pipe flanges shall be not less than that specified by the Manufacturers' Standard for High Pressure.

68. Joint laps shall be not less than one and one-half (1½) times the diameter of the rivet hole.

69. Rivet holes shall be either drilled full size from the solid with plates, butt-straps and heads bolted in position; or they may be punched not to exceed one-fourth (¼) inch less than full size for plates over five-sixteenths ($\frac{5}{16}$) inch in thickness, and one-eighth (⅛) inch less than full size for plates not exceeding five-sixteenths ($\frac{5}{16}$) inch in thickness, and then drilled or reamed to full size with plates, butt-straps and heads bolted up in position.

70. Tube holes shall be drilled full size, or they may be punched not to exceed one-half (½) inch less than full size, and then drilled, reamed or finished full size with rotating cutter, the edges of the holes being chamfered to a radius of about one-sixteenth ($\frac{1}{16}$) inch.

71. A fire-tube boiler shall have the ends of the tubes substantially beaded.

72. The ends of all tubes, suspension tubes and nipples shall be flared not less than one-eighth (⅛) inch over the diameter of the tube hole on all water-tube boilers and superheaters.

73. The ends of all tubes, suspension tubes and nipples of water-tube boilers and superheaters shall not project through the tube sheets or headers less than one-fourth (¼) inch nor more than one-half (½) inch. Separately fired superheaters shall have the tube ends protected by refractory material where they connect with drums or headers.

74. An opening in a boiler for a threaded pipe connection one (1) inch in diameter or over (except water column connections) shall not have less than the minimum number of threads in such opening, as shown in the following table:

THREADS IN PIPE OPENINGS.

Size of pipe connection. Ins.	1 and 1¼	1½ and 2	2½ to 4 inclusive	4½ to 6 inclusive	7 and 8	9 and 10	12
Number of threads per inch.	11½	11½	8	8	8	8	8
Minimum number of threads required in opening.	4	5	7	8	10	12	13
Minimum thickness of material required to give above number of threads. Ins.	0.348	0.435	0.875	1.000	1.250	1.500	1.625

If the thickness of the material in the boiler is not sufficient to give such number of threads, there shall be a standard commercial pressed steel flange, bronze composition flange, cast steel flange or steel plate, substantially riveted to the boiler, so as to give the required number of threads. A main steam or safety valve opening may be fitted with either a cast steel, cast iron or bronze composition nozzle, and a feed-pipe connection may be fitted with a brass or steel boiler bushing.

75. An elliptical manhole opening shall be not less than eleven (11) by fifteen (15) inches in size.

A circular manhole opening shall be not less than fifteen (15) inches in diameter.

76. There shall be a manhole in the upper part of the shell or head of a fire-tube boiler over forty (40) inches in diameter, except a vertical fire-tube boiler.

77. A manhole frame shall be of wrought or cast steel, and shall have a net cross-sectional area, on a line parallel to the axis of the shell, not less than the cross-sectional areas of shell plate removed on the same line.

78. Manhole frames on shells or drums shall have the proper curvature, and on boilers over forty-eight (48) inches in diameter shall be riveted to the shell or drum with two rows of rivets. The strength of the rivets in shear shall not be less than the tensile strength of the shell plate removed, on a line parallel to the axis of the shell, through the center of the manhole.

79. Manhole covers shall be of wrought or cast steel.

80. A manhole shall be located in the front head, below the tubes, of a horizontal return tubular boiler sixty (60) inches or over in diameter.

81. A manhole or handhole shall be located in the front head, below the tubes, of a horizontal return tubular boiler less than sixty (60) inches in diameter.

82. A handhole shall be located in the rear head of a horizontal return tubular boiler, below the tubes, except one which has a manhole in the front head, below the tubes.

83. A locomotive type boiler shall have not less than six (6) handholes, located as follows:

One (1) in the rear head, below the tubes.

One (1) in the front head, at or about the line of the crown sheet.

Four (4) in the lower part of the waterleg.

Also, where possible, one (1) near the throat sheet.

84. A vertical fire-tube boiler, except the boiler of a steam fire engine, shall have not less than four (4) handholes, located as follows:

One (1) in the shell at or about the line of the crown sheet.

One (1) in the shell at or about the line of the fusible plug, except a vertical fire-tube boiler having a manhole in the shell or head, through which the fusible plug is accessible.

Two (2) in the shell at the lower part of the waterleg.

85. A vertical fire-tube boiler of a steam fire-engine shall have not less than three (3) brass washout plugs of not less than one (1) inch pipe size, screwed into the shell and located as follows:

One (1) at or about the line of the crown sheet.

Two (2) at the lower part of the waterleg.

86. There shall be not less than one (1) inch of solid plate in the clear, inside and out, around a handhole opening in a boiler; except that on a staybolted surface seven-sixteenths ($\frac{7}{16}$) inch thick or over there shall be not less than one-half ($\frac{1}{2}$) inch of solid plate in the clear, inside and out, around a handhole opening.

87. A horizontal return tubular boiler over seventy-eight (78) inches in diameter shall be supported, independent of the boiler side-walls, from steel lugs by the outside suspended type of setting; where three (3) supports are necessary on each side of a boiler, an equalizer shall be used.

88. A horizontal return tubular boiler over fifty-four (54) inches in diameter, and up to and including seventy-eight (78) inches in diameter, shall be supported by the outside suspended type of setting, or by not less than four (4) steel or cast iron brackets on each side, set in pairs.

89. A horizontal return tubular boiler up to and including fifty-four (54) inches in diameter shall be supported by the outside suspended type of setting, or by not less than two (2) steel or cast iron brackets on each side.

90. Supporting lugs or brackets shall have the proper curvature and be securely riveted to the shell. The shearing stress on the rivets shall not exceed eight (8) per cent of the ultimate shearing strength given in Part II of these rules.

91. The upper surface of the fire-grate of an internally fired boiler of the open bottom locomotive, vertical fire-tube or similar type, shall not be below the water space in the waterleg, except where the rivets at the bottom of the waterleg are protected from the action of the fire and products of combustion.

92. Wet-bottom boilers shall have a clear space of not less than twelve (12) inches between the bottom of the boiler and the floor line.

93. The feed water shall discharge above the tubes, about three-fifths ($\frac{3}{5}$) the length of a horizontal return tubular boiler from the front head (except a horizontal return tubular boiler equipped with an auxiliary feed-water heating and circulating device), and at or about the central rows of tubes when the diameter of the boiler exceeds thirty-six (36) inches and the pressure allowed exceeds twenty-five (25) pounds per square inch. The feed pipe shall be carried through the head or shell with a brass or steel boiler bushing (or the opening reinforced), and securely fastened inside the shell above the tubes.

94. Feed water shall not be discharged in a boiler in close proximity to riveted joints in shell or furnace sheets.

95. The maximum size of a surface blow-off pipe shall not exceed one and one-half ($1\frac{1}{2}$) inches, and it shall be carried through the shell or head with a brass or steel boiler brushing, or the opening reinforced.

96. Each boiler shall have a bottom blow-off pipe, fitted with a valve or cock, in direct connection with the lowest water space practicable; the minimum size of pipe and fittings shall be one (1) inch and the maximum size shall be two and one-half ($2\frac{1}{2}$) inches. Globe valves shall not be used.

97. A bottom blow-off pipe shall be protected from the products of combustion by a fire-brick casing, substantial cast iron removable sleeve, or covering of non-conducting material.

98. The minimum size of pipes connecting the water column of a boiler shall be one (1) inch.

99. The steam connections to the water column of a horizontal return tubular boiler shall be taken from the top of shell or the upper part of head; the water connection shall be taken from a point not less than six (6) inches below the center line of the shell.

100. The lowest factors of safety to be used in calculating the maximum allowable pressure on boilers which were in this Commonwealth on or before the enactment of these rules, and which are not Massachusetts Standard, or which have not been passed by joint inspection, if hereafter installed, shall be as follows:

(*a*) Six (6) for boilers the longitudinal joints of which are of lap-riveted construction, and the shells or drums of which *are not* exposed to the products of combustion.

(*b*) Six (6) for boilers the longitudinal joints of which are of lap-riveted construction, and the shells or drums of which *are* exposed to the products of combustion, diameters up to and including thirty-six (36) inches.

(*c*) Eight (8) for boilers the longitudinal joints of which are of lap-riveted construction, and the shells or drums of which *are* exposed to the products of combustion, diameters over thirty-six (36) inches.

(*d*) Four and five-tenths (4.5) for boilers the longitudinal joints of which are of butt- and double-strap construction, age not exceeding ten (10) years.

(*e*) Five (5) for boilers the longitudinal joints of which are of butt- and double-strap construction, age over ten (10) years.

The hydrostatic pressure test on such boilers shall be one and one-half (1½) times the maximum allowable pressure obtained by using the above factors of safety.

14. Lined Tanks. — In order to prevent destructive corrosion on the interior surfaces of tanks and pressure vessels, various linings, more or less insoluble in character, have been employed. The metals most generally used for this purpose are copper, lead and silver. Monel metal, an alloy consisting of 72 per cent nickel, 1.5 per cent iron and 26.5 per cent copper, is also well adapted to form non-corrosive linings for mine and sea pumps.

There are two general methods by which the above linings are applied. First, a liner or sleeve of the non-corrosive material may be accurately shaped and forced to place. Second, an interior coating of the insoluble metal may be brazed or soldered directly to the outer shell. Copper and hard alloys are the materials best fitted for forced liners. There is just enough ductility to such bushings to permit tight fits around pistons and valves and thereby prevent leakage. When cold-rolled there is sufficient density to such metals to resist percolation through their pores at very high pressures. Consequently, copper and alloy liners, forced or rolled to place, frequently form the linings of hydraulic cylinders, serving in a double capacity to prevent both corrosion and leakage. When necessary the ends of such sleeves may be hermetically brazed to the exterior cylinder walls.

Soft linings such as those of lead, silver or thin copper are generally soldered or "sweated" to place. The inside surface of the shell must first be thoroughly cleansed by sand blast or otherwise, and given a preliminary coating of the solder or other flux to be used. The latter process can often be best accomplished by heat applied from the outside of the tank. Next, the lining itself, fashioned to a fair fit and given an exterior coating of flux, is inserted in the tank. Air pressure at 100 pounds per square inch or more is then introduced inside of the lining and instantly causes the latter to take the interior shape of the tank. Rivet heads and over-lapping plates present no difficulty to this process provided the lining is thick enough to stretch over such irregularities without rupture. To be certain that the air pressure is confined within the lining, the latter must be soldered to the inlet pipe. When the outer shell contains uncalked riveted joints there is generally sufficient vent through them to let out the air imprisoned between the lining and shell. When of one piece it may be necessary to pierce the outer shell by small holes for the above purpose.

It next remains to heat the tank from without to a degree sufficient to insure the complete union of the shell and lining. The desired temperature can best be determined by noting the time at which a piece of the flux-metal fuses upon the outside of the tank thus indicating the necessary degree of heat within and preventing injury to the lining. Such tanks are of great service in the processes of refrigeration, rendering and reclaiming, as well as for many purely chemical operations carried on in the arts.

CHAPTER II.

STRESSES IN PRESSURE VESSELS.

THEORETICAL formulæ for the determination of stresses in vessels subjected to pressure may, in general, be grouped under three heads, 1°, Spheres, 2°, Cylinders, and 3°, Flat plates. Under spheres should be included the segments of spheres as used for the ends of cylindrical pressure vessels. Spheres and cylinders may have thin or thick shells and be subjected to either internal or external pressure or both. Flat plates may be of different shapes, either supported or fixed at the edges, or the plate may be continuous and supported at a system of points.

Stress formulæ may be derived from a purely theoretical discussion, or the form of the equation, in so far as the variables are concerned, may be obtained from such a discussion and used in connection with an experimental constant. The practice of determining dimensions from a purely empirical equation, based on designs which have not failed, is not to be recommended. In any event the designer must appreciate the assumptions upon which his formulæ are based, and the limits within which they may be used with a reasonable degree of assurance.

Since formulæ may be deduced with reference to either *apparent* or *true stresses*, a clear understanding of these terms is essential. Throughout the following discussion the term stress will be taken to mean an internal force. That is, if a body be subjected to external forces or loads, the stress is that force which particles of the body on one side of a cutting plane exert upon the particles on the other side. Strain is the deformation of the body due to the application of external forces. An apparent stress is one determined by the principles of statics, the body being assumed to be perfectly inelastic. The portion of the body on one side of the plane at which the stress is desired is taken as rigid. The unknown internal force or forces at the plane form part of the system of forces acting on this portion and may be found by applying the conditions of equilibrium. The criterion of safety in this case is that the working stress shall not exceed a certain fraction of the ultimate strength of the material. Expressed algebraically

$$f_{\text{working}} \leqq \frac{\text{Ultimate strength}}{\text{Factor of safety}}.$$

A true stress is one which corresponds to a strain. The term true stress will be used for convenience. It must be borne in mind, however, that such a stress is not a stress in the sense of the definition given above. True stresses are those which would be required to produce the given strains, provided the elastic limit of the material be not exceeded. The elastic properties of the body must be considered in the determination of such a stress. A design based on true stresses is in reality a design of limited strains. In such cases the strain should never exceed that at the elastic limit divided by a factor of safety, or,

$$f_{\text{working}} \leqq \frac{\text{Elastic limit}}{\text{Factor of safety}},$$

where f is a true stress.

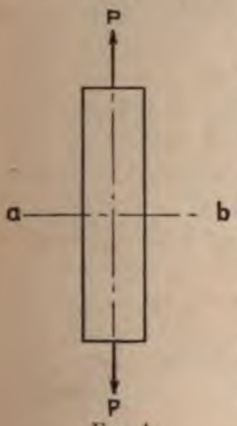


Fig. 4.

Either an apparent or a true stress may be the larger, and care should be taken in any case that both are considered. If they are not, an apparent factor of safety may be considerably in excess of the actual factor. An intelligent choice of constants and factors of safety will, in many cases, give results closely in accord, from equations which seemingly give widely varying results where the same constants are used in all cases. To illustrate the above assume a rectangular prism, Fig. 4, subjected to the external vertical forces P at its ends. These forces cause, at the cross section ab, an apparent, and in this case a true, intensity of stress p. While there is no apparent stress in any horizontal direction there is a strain and hence a true stress in all horizontal directions. This true stress may be obtained from the apparent stress by multiplying by the coefficient of lateral contraction or Poisson's ratio. This ratio is usually taken as $\frac{1}{3}$ or $\frac{3}{10}$ for wrought iron or steel and $\frac{1}{4}$ for cast iron. Final equations are but little affected by the choice of this ratio, and it will be taken as $\frac{1}{3}$ in all cases. The reciprocal of this ratio is designated by "*m*," hence $m = 3$. It is to be noted that an apparent stress of tension in one direction causes a true stress of compression in all directions at right angles. Assume the prism of Fig. 4 to be subjected also to compressive horizontal external forces P', Fig. 5. These forces cause an apparent intensity of stress, p', on a plane perpendicular to *ab*. While the apparent intensity of stress on *ab* is still p, the true intensity of stress is $p + \frac{p'}{3}$, tension. Hence a consideration of both stresses is necessary in an economical design.

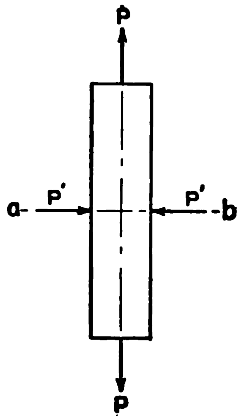


FIG. 5.

The obtaining of true stresses from apparent stresses is not to be confused with the compounding of apparent stresses. In this case apparent stresses on two planes through a given point are used to determine an apparent stress on any other plane through the point. Use will be made of this proposition in connection with helical riveting. Given the point O, Fig. 6, in a body, through which pass planes XX and YY perpendicular to the plane of the

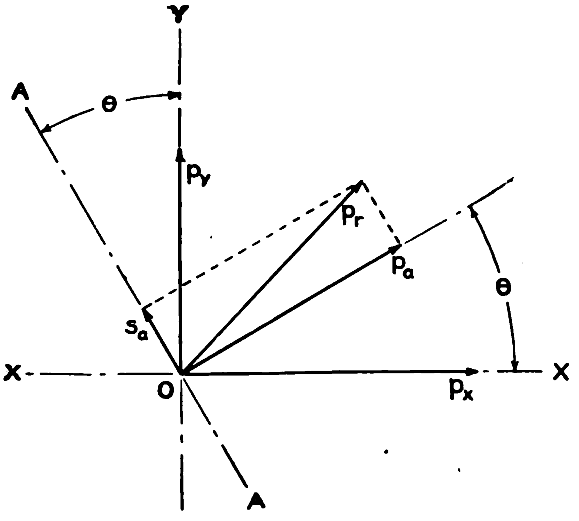


FIG. 6.

paper. Assume as the simplest case, and the only one which it is necessary to consider in this work, that these are Principal Planes, that is, planes upon which the resultant stress is wholly normal. Let these resultant stresses, called Principal Stresses, be denoted by p_x and p_y, unit stresses in each case. The resultant unit stress p_r, on any other plane AA, passing through O, whose normal makes an angle θ with the X axis, is obtained as follows: The intensity of stress on AA due to p_x will be $p_x \cos \theta$ which may be resolved into two components normal and tangential to plane AA.

$$\text{Normal component} = p_x \cos^2 \theta.$$
$$\text{Tangential component} = p_x \cos \theta \sin \theta.$$

Similarly for p_y.

$$\text{Normal component} = p_y \sin^2 \theta.$$
$$\text{Tangential component} = p_y \sin \theta \cos \theta.$$

Hence p_a, the resulting normal component on AA, will equal $p_x \cos^2 \theta + p_y \sin^2 \theta$, and S_a, the tangential component, will equal $(p_x - p_y) \sin \theta \cos \theta$.

Therefore,

$$\begin{aligned} p_r &= \sqrt{p_a^2 + S_a^2} \\ &= [p_x^2 \cos^4 \theta + 2\, p_x p_y \cos^2 \theta \sin^2 \theta + p_y^2 \sin^4 \theta \\ &\qquad + p_x^2 \sin^2 \theta \cos^2 \theta - 2\, p_x p_y \sin^2 \theta \cos^2 \theta \\ &\qquad + p_y^2 \sin^2 \theta \cos^2 \theta]^{\frac{1}{2}} \\ &= [p_x^2 \cos^2 \theta (\cos^2 \theta + \sin^2 \theta) + p_x^2 \sin^2 \theta (\cos^2 \theta + \sin^2 \theta)]^{\frac{1}{2}} \\ &= [p_x^2 \cos^2 \theta + p_y^2 \sin^2 \theta]^{\frac{1}{2}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1) \end{aligned}$$

which is the expression for the resultant stress on plane AA, due to the given principal stresses.

It is evident that p_r will always have values between p_x and p_y, except when $p_x = p_y$, in which case $p_r = p_x = p_y$. Hence one of the principal stresses is a maximum and the other a minimum.

15. Thin Hollow Sphere. — When a spherical shell is so thin that the stress may be taken as uniformly distributed from the inside wall to the outside wall, the calculation of this stress is a comparatively easy matter.

Let Fig. 7 represent the upper half of a thin sphere, where

r = internal radius of sphere,
t = thickness of shell,
p = intensity of internal pressure,
f = intensity of tangential stress (hoop tension).

Considering this portion of the sphere as a solid body, the total force tending to raise it is $\pi r^2 p$. This must be equal to the stress with which the lower portion acts upon the upper portion at the section, or $2\pi rft$, very nearly.

Hence,
$$\pi r^2 p = 2\pi rft$$
$$f = \frac{pr}{2t}. \qquad (2)$$

In this case f is, of course, only an apparent principal stress, and in any given case would be taken as the ultimate tensile strength of the material with a proper factor of safety.

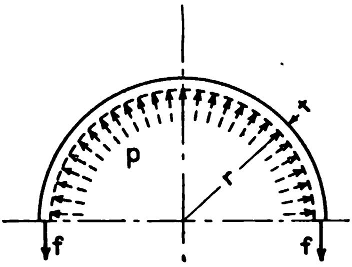


FIG. 7.

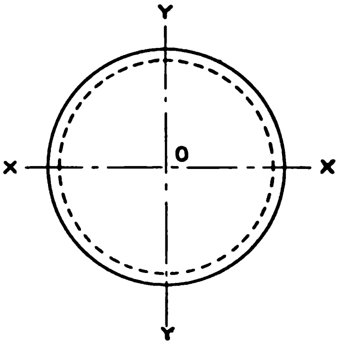


FIG. 8.

To illustrate the design on the basis of maximum strains, consider the point O lying in the wall of the spherical shell, Fig. 8, at the intersection of planes XX and YY. There is an intensity of stress normal to plane YY, $f_y = \frac{pr}{2t}$, from equation (2). There is also an intensity of stress normal to plane XX, $f_x = \frac{pr}{2t}$. The radial stress p, also acting, if O is on the inside surface of the wall, is neglected. The stresses f_x and f_y are apparent stresses from which the true tangential stress may be found thus:

$$f_{\text{true}} = f_x - \tfrac{1}{3} f_y = f_y - \tfrac{1}{3} f_x$$
$$= \frac{pr}{2t} - \frac{pr}{6t}$$
$$= \frac{pr}{3t}. \qquad (3)$$

Thus it is seen that the true stress is only two-thirds of the apparent stress. The design should be such that the value of this true stress falls well within the elastic limit of the material.

Equations (2) and (3) hold equally true for thin hollow spheres subjected to external pressure, but they are of little value when so used, because slight deviations from the true spherical form are rapidly exaggerated, and the sphere fails by collapse rather than by direct compression.

16. Thick Hollow Sphere. — It is desired to obtain the intensity of hoop tension at the inner surface of the thick hollow sphere, Fig. 9, subjected to internal pressure only.

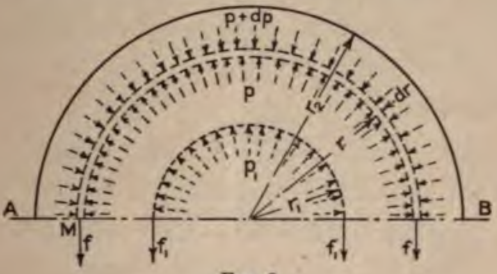


FIG. 9.

Let f_1 = intensity of hoop tension.
r_1 = internal radius of sphere.
r_2 = external radius of sphere.
p_1 = intensity of internal pressure.
r = radius of an elementary annulus.
p = radial pressure on the elementary annulus (inside).
$p + dp$ = radial pressure on the elementary annulus (outside).
f = intensity of tangential stress on annulus.

A small particle at M is subjected to a stress f, normal to the horizontal plane AB, a stress f, normal to the plane of the paper, and a radial stress p. It can be shown by the principle of least work that the algebraic sum of these stresses is a constant for all points in the sphere. Hence,

$$2f - p = 2f_1 - p_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

tensions being written positive and compressions negative. Consider the forces acting on the elementary hemispherical annulus,

Acting upward, $\pi r^2 p$.
Acting downward, $\pi (r + dr)^2(p + dp)$ and $2\pi r f\, dr$.
Equating, $\pi r^2 p = \pi (r + dr)^2(p + dp) + 2\pi r f\, dr$.

Solving, omitting differentials of more than the first power,

$$r\,dp = -2p\,dr - 2f\,dr,$$

$$\frac{dp}{p+f} = -\frac{2\,dr}{r}.$$

From equation (4) $f = \dfrac{p + 2f_1 - p_1}{2}.$

Therefore,

$$\frac{2\,dp}{3p + 2f_1 - p_1} = -\frac{2\,dr}{r},$$

or

$$\frac{3\,dp}{3p + 2f_1 - p_1} = -\frac{3\,dr}{r}.$$

Hence, by integration,

$$\log(3p + 2f_1 - p_1) = -3\log r + \log c,$$

where log c is the constant of integration.

$$\log(3p + 2f_1 - p_1) + \log r^3 = \log c,$$

$$\log[(3p + 2f_1 - p_1)(r^3)] = \log c,$$

or

$$3p + 2f_1 - p_1 = \frac{c}{r^3}. \qquad (5)$$

$$p = \frac{1}{3}\left(p_1 - 2f_1 + \frac{c}{r^3}\right). \qquad (6)$$

To find c, note that, when $r = r_2$, $p = 0$.

$$c = (2f_1 - p_1)\,r_2^3.$$

Then, substituting for p in equation (5) its equal from equation (4), $2f + p_1 - 2f_1$,

$$6f + 3p_1 - 6f_1 + 2f_1 - p_1 = \frac{r_2^3}{r^3}(2f_1 - p_1),$$

$$6f = 4f_1 - 2p_1 + \frac{r_2^3}{r^3}(2f_1 - p_1),$$

which is the expression for the tangential stress at any radius r.

When $r = r_1$, $f = f_1$.

Hence,

$$6f_1 = 4f_1 - 2p_1 + \frac{r_2^3}{r_1^3}(2f_1 - p_1),$$

and

$$f_1 = \frac{p_1\left(\dfrac{r_2^3}{2} + r_1^3\right)}{r_2^3 - r_1^3}, \qquad (7)$$

or

$$r_2 = r_1\sqrt[3]{\frac{2f_1 + 2p_1}{2f_1 - p_1}}. \qquad (8)$$

Equation (7) then gives the apparent intensity of hoop or tangential tension at the inside surface of a thick spherical shell, this being the maximum stress. Stresses calculated by equation (7) will always be in excess of those calculated by equation (2) for thin spheres. The error, however, will be only six-tenths of one per cent where the thickness is not more than ten per cent of the internal radius, and not over two per cent when the internal pressure does not exceed one-third of the allowable fibre stress.

The true intensity of stress at the inside surface is found from the coefficient of lateral contraction, thus:

$$f_{\text{true}} = f_1 - \frac{f_1}{3} + \frac{p_1}{3}$$

$$= \frac{p_1}{3}\left(\frac{2\,r_2^3 + r_1^3}{r_2^3 - r_1^3}\right). \quad \ldots\ldots \quad (9)$$

or

$$r_2 = r_1\sqrt[3]{\frac{3\,f_{\text{true}} + p_1}{3\,f_{\text{true}} - 2\,p_1}}. \quad \ldots\ldots \quad (10)$$

Evidently f_{true} will be less than f_1 for all cases where p_1 is less than f_1, that is when the internal pressure does not exceed the allowable fibre stress. It is interesting to note that for cases where the above is not true, equation (7) for the apparent stress is on the unsafe side, and equation (9) should be used. Such a case might be that of a cast iron sphere subjected to an internal pressure of 5000 pounds per square inch with an allowable working fibre stress of 4500 pounds per square inch.

17. Thin Hollow Cylinder. — Vessels of a cylindrical shape are the ones most widely used to withstand fluid pressures. Where the thickness of the cylinder is small compared with the diameter and the pressure is internal, the tensile stress in the material may be computed readily and with considerable assurance. The same cylinder subjected to external pressure, as is the case in a boiler tube, may have the stresses computed in a similar manner. The slightest deviation from the true cylindrical form, however, is soon increased by the pressure and failure occurs by the collapsing of the tube, thus rendering any calculations for failure by compression valueless. In these cases it is necessary to resort to tests to obtain collapsing pressures. The results of such tests will be given later. To obtain the intensity of tangential stress in a thin hollow cylinder due to a given working pressure, take a section of unit length, Fig. 10, and let

r = internal radius of cylinder,
t = thickness of metal,
p = intensity of internal pressure,
f = intensity of hoop tension, assumed constant across the section.

The pressure acting within the upper half of the cylinder tends to cause a separation on the horizontal plane and is resisted by the stresses in the material on this plane. While all pressures within the cylinder act normally, the sum of the vertical components of all such forces will be equal in magnitude to the resultant force on the plane *abcd*. The latter is equal to 2 rp, and is resisted by the stresses on the two sections of the material, each of which is ft, assuming the metal so thin that the stress is uniformly distributed. The stress f may be taken as the maximum stress, although it is in reality slightly less than the maximum.

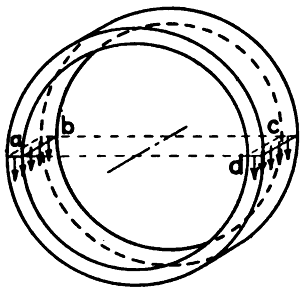


FIG. 10.

Equating, $$2\,rp = 2\,ft,$$

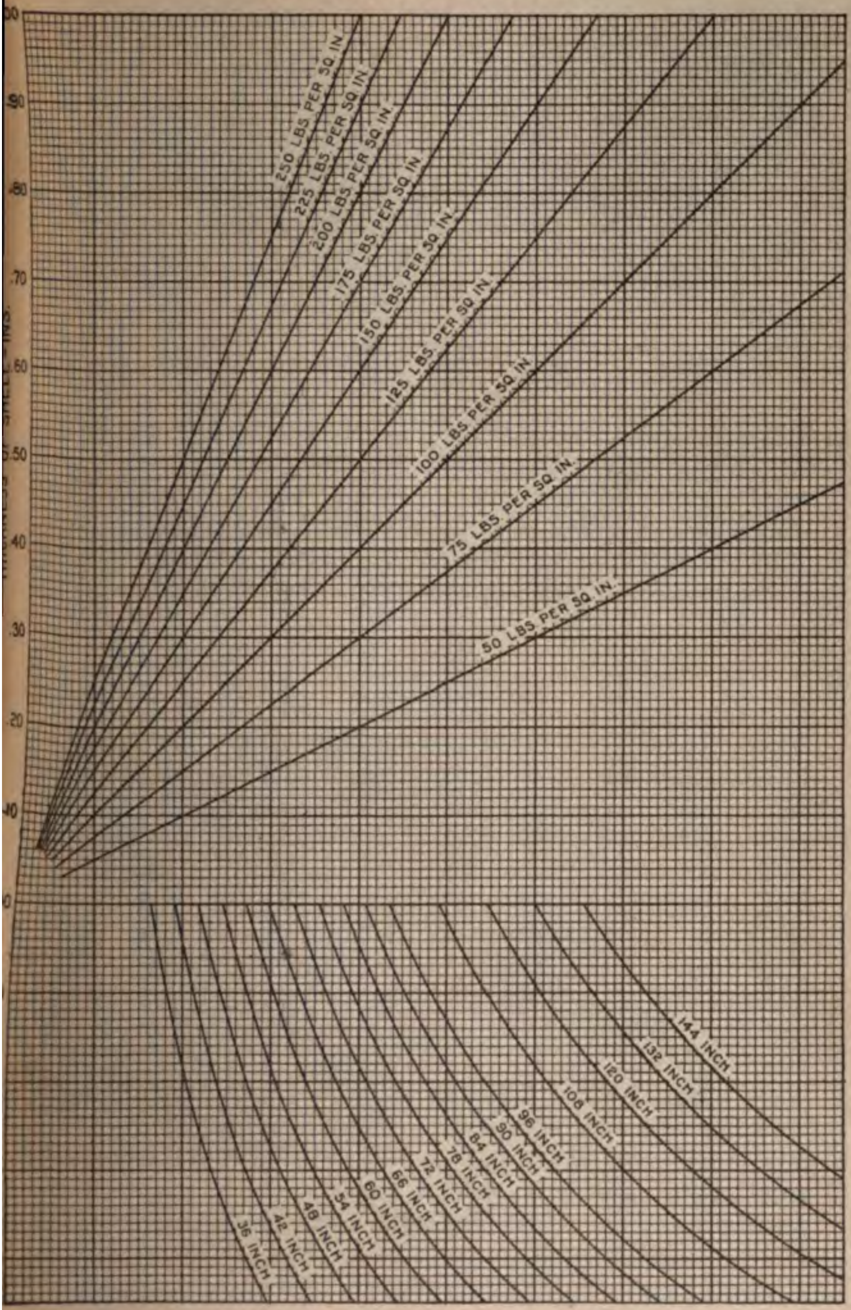
and $$f = \frac{pr}{t},$$

or $$t = \frac{pr}{f}. \qquad \qquad (11)$$

Where it is necessary to use a longitudinal joint whose efficiency is e, the equation becomes

$$t = \frac{pr}{fe}. \qquad \qquad (12)$$

The plot, Fig. 11, has been constructed for a ready determination of thicknesses as obtained from equation (12). f is taken as 55,000 ÷ 5 = 11,000 pounds per square inch. To use the plot, follow the horizontal line corresponding to the efficiency of the joint to its intersection with the curve corresponding to the cylinder diameter. Rise vertically to the line of the desired internal pressure. Read the thickness of plate corresponding at the left.



CYLINDRICAL SHELL THICKNESSES

Fig. 11.

Problem.

Find the necessary thickness of shell for a 72 inch boiler, 150 lbs. per sq.-in. pressure, having a joint with an efficiency of 88 per cent.

From plot $t = 0.556$ in. Use $\frac{9}{16}$ inch.

The thickness necessary to withstand the pressure on the end of the cylinder may be computed by considering a transverse cross section. The total load imposed upon the section is $\pi r^2 p$ The resisting stress is $2\pi r f t$, very nearly.

Hence,
$$f = \frac{pr}{2t}$$

or
$$t = \frac{pr}{2f}. \qquad (13)$$

This equation is the same as that obtained for thin hollow spheres. With a given thickness of shell the value of the hoop tension is twice that of the longitudinal tension. Hence, a girth seam joint of more than one-half the efficiency of the longitudinal joint will render the vessel safe against rupture in this direction.

The true intensity of hoop tension, f_{true}, is equal to

$$f - \frac{1}{3}\cdot\frac{f}{2} + \frac{p}{3}.$$

Therefore $f_{\text{true}} = \frac{5}{6} f$ (nearly), f being large compared with p. Hence, designs based on the apparent stress are on the safe side. It must be borne in mind, however, that where the principal stresses are of opposite kinds, the true stress exceeds the apparent stress, and should be used in design. As an example of such a case, consider a stand-pipe, where the normal stress on a vertical longitudinal section is tension due to liquid pressure, and the stress on a horizontal transverse section is compression. Then the maximum intensity of true stress is, $f_{\text{true}} = f_1 + \frac{1}{3} f_2$ where f_1 and f_2 are the principal stresses with f_1 larger than f_2, and f_{true} is tension or compression according as f_1 is tension or compression.

18. Thick Hollow Cylinder. — There are two assumptions used in determining equations for the stresses in thick cylinders. The simpler one, due to Barlow, is that the area of a transverse cross section remains a constant whatever the internal and external pressures may be. The other, developed by Lamé, is that the longitudinal tensile stress and longitudinal elongation are constant for all points in the cylinder. The latter assumption is the more exact, but the Barlow equation gives results on the safe side in all cases, as is seen by the plot, Fig. 13. It is necessary in both cases to assume the length of the cylinder large compared with the outside diameter, in order that the influence of the ends shall not affect the reasoning.

Deduction of stress equation on first assumption.

Referring to Fig. 12.

Let r_1 = internal radius of cylinder before pressure is applied.
r_2 = external radius of cylinder before pressure is applied.
p_1 = intensity of internal pressure.
p_2 = intensity of external pressure.
f_1 = intensity of hoop tension at inner surface.
f_2 = intensity of hoop tension at outer surface.
x_1 = increase in r_1 after pressure is applied.
x_2 = increase in r_2 after pressure is applied.

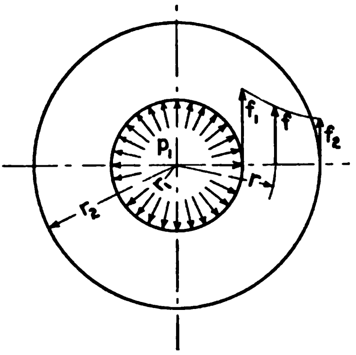


FIG. 12.

Area of transverse cross section, before applying pressure

$$\pi\,(r_2^2 - r_1^2). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Area of transverse cross section, after applying pressure

$$\pi\,[(r_2 + x_2)^2 - (r_1 + x_1)^2]. \quad . \quad . \quad . \quad . \quad . \quad (15)$$

By assumption (14) = (15).

Therefore, $r_2^2 - r_1^2 = r_2^2 + 2\,r_2x_2 + x_2^2 - r_1^2 - 2\,x_1r_1 - x_1^2$.

Neglecting the terms x_2^2 and x_1^2, since they are extremely small and appear in the equation with opposite signs

$$r_2x_2 = r_1x_1,$$

or

$$\frac{x_1}{x_2} = \frac{r_2}{r_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Unit strain in the internal circumference will equal

$$\frac{2\,\pi\,(r_1 + x_1) - 2\,\pi r_1}{2\,\pi r_1} = \frac{x_1}{r_1}.$$

Similarly for the external circumference the unit strain will equal

$$\frac{x_2}{r_2}.$$

Hence,

$$\frac{\text{Strain at inner circumference}}{\text{Strain at outer circumference}} = \frac{\dfrac{x_1}{r_1}}{\dfrac{x_2}{r_2}} = \frac{f_1}{f_2}.$$

$$\frac{f_1}{f_2} = \frac{x_1 r_2}{x_2 r_1} = \frac{r_2^2}{r_1^2},$$

or the hoop tensions vary inversely as the squares of the distances of the fibres from the center of the cylinder. Professor Goodman, in a series of experiments, found the stresses over a cross section to vary very closely as given by this relation. If f is the intensity of hoop tension at any point in the cross section at a distance r from the center of the cylinder, the total stress, P over the cross section for unit length of section will be

$$P = \int_{r_1}^{r_2} f\,dr,$$

but, as shown above,

$$f = f_1 \frac{r_1^2}{r^2}.$$

Therefore,

$$P = f_1 r_1^2 \int_{r_1}^{r_2} \frac{dr}{r^2}.$$

$$= f_1 r_1 - f_1 \frac{r_1^2}{r_2}$$

$$= f_1 r_1 \left(1 - \frac{r_1}{r_2}\right).$$

Applying the conditions of equilibrium to the half shell of unit length, and neglecting p_2 as small compared with p_1,

$$2\,r_1 p_1 = 2\,P,$$

$$p_1 r_1 = f_1 r_1 \left(1 - \frac{r_1}{r_2}\right),$$

$$f_1 = \frac{p_1 r_2}{r_2 - r_1} = \frac{p_1 r_2}{t}. \qquad (17)$$

This equation is identical with that obtained for a thin shell, except that the outside radius is used in place of the inside radius.

It is to be noted that this expression is for the apparent stress, and not for the true stress.

The deduction of the stress equation under the second or Lamé assumption is entirely analogous to that already given for thick hollow spheres. Therefore, only the results will be given here. The pressure on the outside of the cylinder is assumed to be zero. Then f_1, the apparent maximum intensity of hoop tension at the inner surface, is given by the equation

$$f_1 = \frac{p_1 (r_2^2 + r_1^2)}{r_2^2 - r_1^2}. \qquad (18)$$

For the outer surface,

$$f_2 = \frac{2 p_1 r_1^2}{r_2^2 - r_1^2}.$$

An equation involving thickness may be obtained from equation (18) as follows:

$$\frac{f_1}{p_1} = \frac{r_2^2 + r_1^2}{r_2^2 - r_1^2}.$$

By composition and division,

$$\frac{f_1 + p_1}{f_1 - p_1} = \frac{2 r_2^2}{2 r_1^2}.$$

$$r_2 = r_1 \sqrt{\frac{f_1 + p_1}{f_1 - p_1}}.$$

Subtracting r_1 from both sides,

$$r_2 - r_1 = r_1 \left[\sqrt{\frac{f_1 + p_1}{f_1 - p_1}} - 1 \right].$$

Hence,

$$t = r_1 \left[\sqrt{\frac{f_1 + p_1}{f_1 - p_1}} - 1 \right].$$

When the external pressure, p_2, is *not* neglected the general stress equation is

$$f = \frac{p_1 r_1^2 - p_2 r_2^2 + \frac{r_1^2 r_2^2}{r^2} (p_1 - p_2)}{r_2^2 - r_1^2}. \qquad (19)$$

This equation for apparent stress was modified by Clavarino, who obtained an equation for the true stress,

$$f_{true} = \frac{p_1 r_1^2 - p_2 r_2^2 + \frac{4 r_1^2 r_2^2}{r^2} (p_1 - p_2)}{3 (r_2^2 - r_1^2)},$$

or when $r = r_1$ and $p_2 = 0$,

$$f_{true} = \frac{p_1 (4 r_2^2 + 2 r_1^2)}{3 (r_2^2 - r_1^2)}. \qquad (20)$$

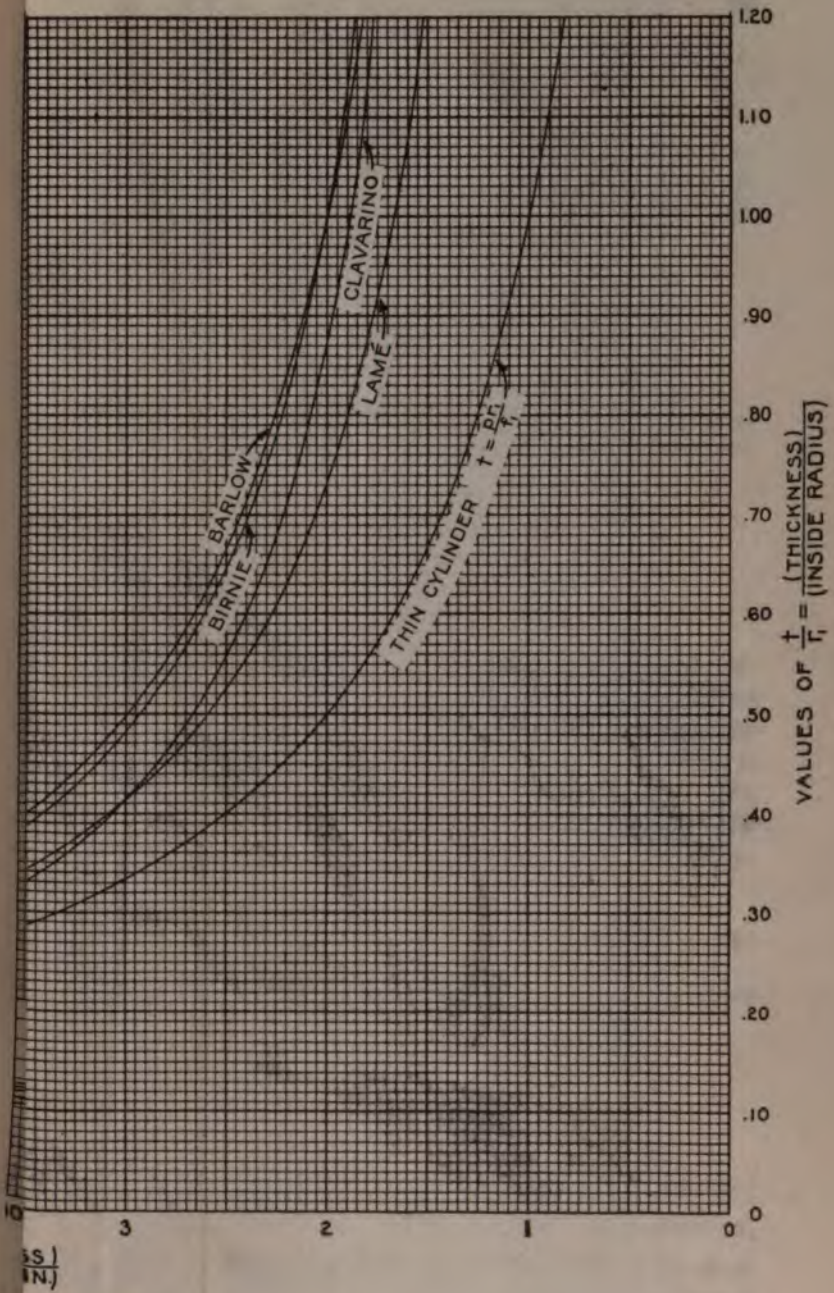
Another modification of Lamé's formula is that due to Bir and is applicable where the longitudinal tension is zero, as in case of a shrink or force fit. Then

$$f_{\text{true}} = \frac{2\,p_1 r_1^2 - 2\,p_2 r_2^2 + \dfrac{4\,r_1^2 r_2^2}{r^2}(p_1 - p_2)}{3\,(r_2^2 - r_1^2)},$$

or, when $r = r_1$ and $p_2 = 0$,

$$f_{\text{true}} = \frac{p_1\,(4\,r_2^2 + 2\,r_1^2)}{3\,(r_2^2 - r_1^2)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2$$

The plot, Fig. 13, has been drawn for the comparison of resul as obtained from equations (17), (18), (20) and (21) and to prese a ready means of solution of problems on thick cylinders. R sults as obtained from equation (11) for thin cylinders are als given to show the limitations of the assumption of uniforml distributed stress. Values of the ratio $\frac{f_1}{p_1}$, that is allowable fib stress to internal pressure, are plotted as ordinates on values the ratio $\frac{t}{r_1}$ as abscissæ. It is to be noted that for ratios of greater than 10, that is when the thickness is less than $\frac{1}{10}$ the i side radius, the equation for thin cylinders gives results in ve close agreement with those of the more complex theories. gives, indeed, thicknesses which are greater than those obtaine from the formula of Clavarino for all values of $\frac{f_1}{p_1}$ greater than When the internal pressure is more than one-third of the allo able working fibre stress, the Lamé formula appears to give r sults on the unsafe side, that is the apparent stress is less tha the true stress. The choice of a factor of safety and of a stre formula affects the resulting thickness of shell very materiall where the internal pressure is more than four-tenths of the allo able working stress. For example, a cast iron cylinder 5 inch inside radius, subjected to 2000 pounds per square inch intern pressure, would have a thickness of 2.83 ins., using the Lam formula, a tensile strength of 19,000 lbs. per sq. in. and a factor safety of 4. The same cylinder would have a thickness of 5.51 in when calculated with the Barlow formula and a factor of 5. Th thickness approaches infinity as $\frac{f_1}{p_1}$ approaches one, in the Barlo


BARLOW
BIRNIE
CLAVARINO
LAMÉ
THIN CYLINDER $t = \frac{pr_1}{f_1}$
3
2
1
0
SS)
IN.)
0
.10
.20
.30
.40
.50
.60
.70
.80
.90
1.00
1.10
1.20
VALUES OF $\frac{t}{r_1} = \frac{\text{(THICKNESS)}}{\text{(INSIDE RADIUS)}}$

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and Lamé formulæ, and as this ratio approaches one and one-third in the Clavarino and Birnie formulæ.

Problems.

(1) Find the necessary thickness of a cylinder 10 inches inside diameter subjected to an internal fluid pressure of 6000 pounds per square inch. Material, steel casting, $f_t = 60,000$ pounds per sq. inch. Factor of safety, 5.

$$\frac{f_1}{p_1} = \frac{12,000}{6,000} = 2.$$

From plot (Clavarino) $\frac{t}{r_1} = 0.872.$

$$t = 5 \times 0.872 = 4.36 \text{ ins.}$$

Lamé curve gives $t = 3.66$ ins.

Barlow curve gives $t = 5.00$ ins.

(2) Find the allowable pressure in a hydraulic main 4 inches in diameter and 1 inch thick, if the stress is not to exceed 5000 pounds per square inch.

$$\frac{t}{r_1} = \frac{1}{2}.$$

From Clavarino curve $\frac{f_1}{p_1} = 2.66,$

$$p_1 = \frac{5000}{2.66} = 1880 \text{ lbs. per sq. inch,}$$

or using Barlow curve $p_1 = 1670$ lbs. per sq. inch.

19. Cylinders Under External Pressure. — Theoretical formulæ for the stresses in cylinders subjected to external pressure are of little value, since such cylinders will collapse before the compressive strength of the material is reached. Especially is this true where the cylinders are thin and long compared with the diameter. Practical examples of cylinders so stressed are the furnace flues and tubes of boilers. The stress equation for thin cylinders is obtained as in the case of internal pressure,

$$f = \frac{pr}{t} \quad \text{or} \quad p = \frac{2ft}{d}, \qquad \text{(22)}$$

where f = mean intensity of compressive stress, sensibly equal to the maximum,
p = intensity of external pressure,
r = external radius,
d = external diameter,
t = thickness of metal.

Theoretical discussions by Professor A. E. H. Love, Grashof, and others show the collapsing pressure to vary as $\left(\frac{t}{d}\right)^n$. However, experiments show clearly that the collapsing pressure for short tubes is not independent of the length but increases as the length decreases for tubes in which the ratio of length to diameter is comparatively small. For a given tube diameter a length is reached such that the collapsing pressure is unaffected by a further increase in length. This is called the critical length and will be taken as six diameters although recent experiments * tend to show that it is somewhat greater and that the following formulæ give low collapsing pressures.

For lengths less than six diameters tests by Sir W. Fairbairn and Professor R. T. Stewart suggest the equation, for inch units

$$p = C\frac{t^{2.25}}{ld},$$

where the constant C as determined from the experiments is 11,600,000. With a factor of safety of 4

$$p_{\text{working}} = 2{,}900{,}000\frac{t^{2.25}}{ld}. \quad \ldots\ldots \quad (23)$$

The furnaces of boilers, while falling within this limit of length to diameter, are made so strong against collapse by corrugations that practical formulæ of the form

$$p_{\text{working}} = C\frac{t}{d} \quad \ldots\ldots\ldots \quad (24)$$

are in general use, the constant varying from 10,000 to 18,000.

If the length is greater than six diameters the collapsing pressure is a function of $\frac{t}{d}$ only. The following values are given by Professor Carman as the results of a large number of experiments and are virtually the same as those obtained by Professor R. T. Stewart from the tests on over 500 boiler tubes.

(*a*) Tubes for values of $\frac{t}{d} < 0.025$.

$$\text{Brass } p = 25{,}150{,}000\left(\frac{t}{d}\right)^3. \quad \ldots\ldots\ldots \quad (25)$$

$$\text{Cold drawn seamless steel } p = 50{,}200{,}000\left(\frac{t}{d}\right)^3. \quad \ldots \quad (26)$$

(*b*) Tubes for values of $\frac{t}{d} > 0.03$.

$$\text{Brass } p = 93{,}365\frac{t}{d} - 2474. \quad \ldots\ldots\ldots \quad (27)$$

$$\text{Cold drawn seamless steel } p = 95{,}520\frac{t}{d} - 2090. \quad \ldots \quad (28)$$

$$\text{Lap-welded steel } p = 83{,}270\frac{t}{d} - 1025. \quad \ldots\ldots \quad (29)$$

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* *The Mechanical Engineer*, Oct. 16, 1914.

(c) Tubes where $\frac{t}{d} > 0.02 < 0.06$ covering all common diameters and thicknesses of boiler tubes.

$$\text{Cold drawn seamless steel } p = 1{,}000{,}000\left(\frac{t}{d}\right)^2 . \quad . \quad (30)$$

$$\text{Lap-welded steel } p = 1{,}250{,}000\left(\frac{t}{d}\right)^2 . \quad . \quad . \quad . \quad . \quad (31)$$

The stress equation for thick cylinders subjected to external pressure only may be obtained from equation (19) by substituting $p_1 = 0$ and $r = r_1$, giving

$$f = -\frac{2\,p_2 r_2^2}{r_2^2 - r_1^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

$$= -\frac{2\,p_2 r_2}{t}\left(\frac{r_2}{2\,r_2 - t}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

the minus sign indicating compression.

Where the thickness is not more than one-tenth of the radius, equation (22) may be used without appreciable error.

20. Spherical Heads. — When possible, cylindrical pressure vessels should be supplied with "dished" or "bumped" heads in order to avoid the staying necessary with flat ends. It is recommended, however, that the ends of all cylinders more than 60 ins. in diameter be stayed. These ends are ordinarily spherical segments, and the stresses are calculated by means of the formula for thin spheres, viz.: $f = \frac{pr}{2t}$ or $p = \frac{2ft}{r}$, where r is the radius of the sphere, and p is the pressure on the concave side. Thus it may be seen that where the stress in the end is to be the same as that in the cylindrical shell with the same thickness in both cases, the radius of the sphere should be twice that of the cylinder. When the pressure is on the convex side of the head, federal and state rules require that the allowable pressure p shall not exceed from six- to eight-tenths of that given above.

When the rise of segment h is known, it can readily be shown that $r = \frac{r_1^2 + h^2}{2h}$, where r_1 is the radius of the cylindrical shell. Hence, by substitution, the equation for allowable pressure becomes

$$p = \frac{4\,fth}{r_1^2 + h^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

in which r_1 is the radius of the cylindrical shell. While the above formulæ are those ordinarily used in connection with dished heads, they are far from satisfactory and should only be employed in connection with a large factor of safety.

Stresses in the flange of the head are very severe, but they do not admit of a ready analysis. While they occur at a place of considerable inherent strength due to the joint and the flanging of the head, nevertheless, grooving and severe local bending stresses at the sharp curvature cause cracks and ultimate rupture at this point. Experimental determinations of the strains in the steam drum of a Babcock and Wilcox boiler have been made by Professors S. H. Barraclough and A. J. Gibson and reported in the "Proceedings of the Engineering Association of New South Wales," 1910–1911. The accompanying cut, Fig. 14, is taken from their report. The section CC is located 15 ins. from the nearer ring seam. Their investigation was primarily to obtain the stresses in the vicinity of the lap joints, but strains in the flange of the head were also measured. Strain measurements were taken at frequent intervals on transverse and horizontal planes by means of mirrors. Both change of length and change of curvature were

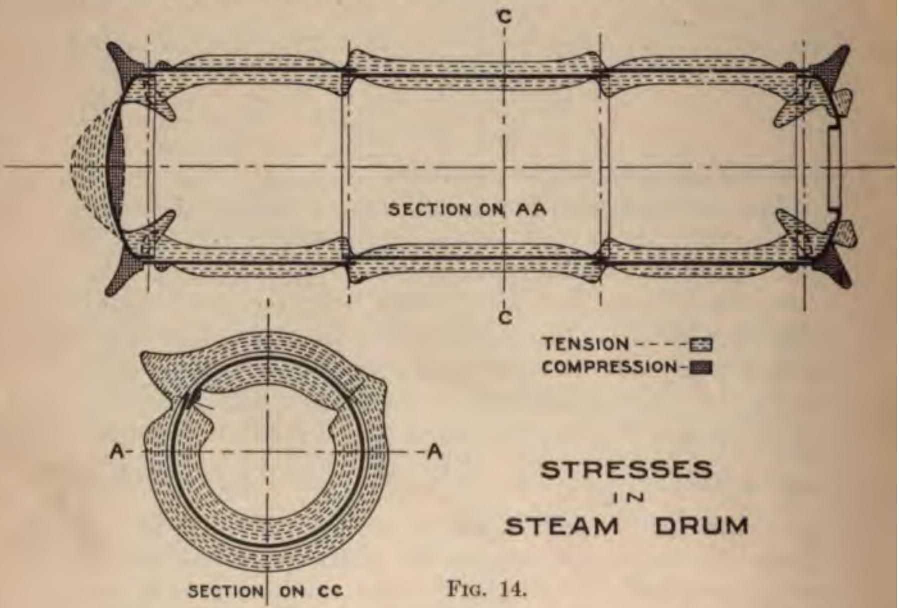
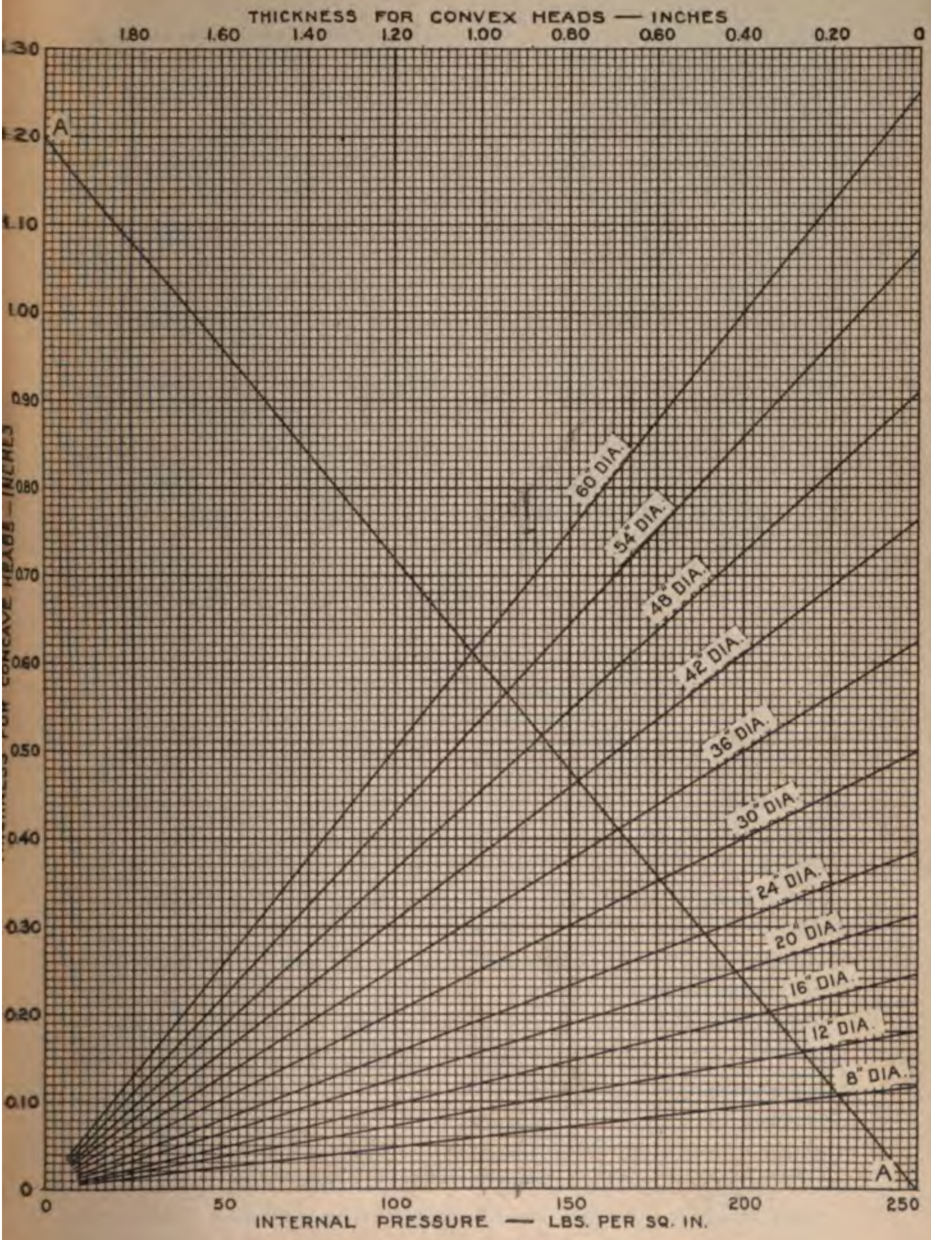


Fig. 14.



THICKNESS OF DISHED HEADS

FIG. 15.

determined. In Fig. 14, "deduced" surface stresses, that is, strains multiplied by a constant (modulus of elasticity) have been plotted perpendicular to the plate, tensions being denoted by dotted and compressions by full lines. The stresses shown in the spherical segment of the head are conjectural, no strains having been measured in this portion. The variation of stress in the vicinity of the single-riveted lap joint is in accord with the usual inferences on this point. The stresses in the flange of the head show a decided bending action with large tensions (11,000 pounds per square inch) on the inside fibres, and compression on the outside fibres. Inasmuch as this is the point at which grooving and rupture take place, the fallacy of using the formula for the thickness of thin spheres, which does not apply to this portion of the head, is apparent. The tensile stress mentioned above is three times that which would have been obtained with the thin sphere formula under the same conditions. The following formula is proposed for the thickness of dished heads,

$$t = \frac{pr}{9000 - 100\,r}, \quad \dots\dots \quad (35)$$

where t = thickness of head in inches.
p = pressure in pounds per square inch.
r = radius of the cylindrical shell in inches.

This formula is intended to apply where the head is dished to a radius equal to the diameter of the shell, and with steel plate of 60,000 pounds per square inch tensile strength. It embodies an apparent factor of safety varying from 7.5 on cylinders 5 inches in diameter, to 10 on cylinders 5 feet in diameter. In the light of the experiments alluded to above it is probable that the true factor of safety for heads calculated by the above formula is in the vicinity of five or six.

When the pressure is on the convex side of the head, thicknesses at least one and two-thirds those given by the above formula are recommended. The plot, Fig. 15, affords a ready means of obtaining the thickness of the head when the diameter of the shell and the internal pressure are known. The thickness for a concave head may be read directly at the left of the plot, and for a convex head by following the abscissa for the thickness of the corresponding concave head to its intersection with line AA and reading the desired thickness at the top of the plot.

It must be borne in mind that the corrosive action of the contents of the tank has much to do with the final thickness of metal used in the heads. The plot gives the theoretical thickness necessary to confine the corresponding pressure. In small cylinders it is frequently best to assume a minimum thickness of head sufficient to withstand safely the corrosion, and at the same time to permit the ready working of the metal to shape. This practical limit will often exceed the theoretical thickness given by the plot.

21. Flat Plates. — The deduction of formulæ for determining the stresses in flat plates supported at their edges and loaded by forces perpendicular to their flat faces forms one of the most difficult chapters in the theory of elasticity. The equations most widely quoted are those of Grashof * deduced from the equation of the elastic curve and giving the maximum true stress, that is the stress corresponding with the strain. Bach adopts equations of the same form, but determines his constants experimentally.

The following cases will be considered:

Case I. Circular plates.
- (*k*) *Uniform load, supported at edges. Fig.* 16.
- (*l*) *Uniform load, fixed at edges. Fig.* 17.
- (*o*) *Uniform load, free at edges and supported at center by a circle of radius* r_0. *Fig.* 18.

- (*x*) *Load* P, *distributed over a circle of radius* r_0, *supported at edges. Fig.* 19.
- (*y*) *Load* P, *distributed over a circle of radius* r_0, *fixed at edges. Fig.* 20.

Case II. Elliptical plates.
- (*k*) *Uniform load, supported at edges. Fig.* 21.
- (*l*) *Uniform load, fixed at edges. Fig.* 22.

- (*x*) *Load* P, *distributed over a circle of radius* r_0, *supported at edges. Fig.* 23.
- (*y*) *Load* P, *distributed over a circle of radius* r_0, *fixed at edges. Fig.* 24.

---

* "Theorie der Elastizität und Festigkeit," 1878, page 329 *et seq.*

Case III. Rectangular plates.

(*k*) *Uniform load, supported at edges.*
(*l*) *Uniform load, fixed at edges. Fig.* 25.
(*x*) *Load P, distributed over a circle of radius* r_0, *supported at edges.*
(*y*) *Load P, distributed over a circle of radius* r_0, *fixed at edges. Fig.* 26.

Case IV. Square plates.

(*k*) *Uniform load, supported at edges.*
(*l*) *Uniform load, fixed at edges. Fig.* 27.
(*x*) *Load P, distributed over a circle of radius* r_0, *supported at edges.*
(*y*) *Load P, distributed over a circle of radius* r_0, *fixed at edges. Fig.* 28.

Case V. Continuous plates.

(*l*) *Uniform load, stayed at a system of points. Fig.* 29.

The significance of the quantities used in the succeeding discussion, when not evident from the illustrations, is as follows:

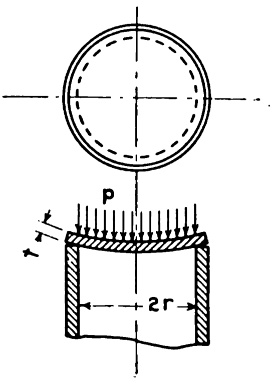


FIG. 16.

f = intensity of apparent stress.
f_{true} = intensity of true stress.
$\frac{1}{m}$ = Poisson's ratio.
v = maximum deflection.
E = modulus of elasticity.
μ = experimental constant.

Case I. Circular plates.

(*k*) *Uniform load, supported at edges. Fig.* 16.

$$f_{\text{true}} = \frac{3}{8}\frac{(m-1)(3m+1)}{m^2}\frac{pr^2}{t^2}.$$

For $m = 3$,

$$f_{\text{true}} = \frac{5pr^2}{6t^2}$$

or

$$t = \sqrt{\frac{5\,pr^2}{6\,f_{\text{true}}}}. \tag{36}$$

$$v = \frac{2\,pr^4}{3\,Et^3}. \tag{37}$$

The maximum apparent stress is

$$f = \frac{3}{8}\left(\frac{3\,m+1}{m}\right)\frac{pr^2}{t^2}.$$

For $m = 3$,

$$f = \frac{5\,pr^2}{4\,t^2}, \tag{38}$$

which is larger than the true stress since the apparent stresses at right angles in the plate are of the same kind.

Bach gives

$$f = \mu\frac{pr^2}{t^2}, \tag{39}$$

where μ is an experimental constant varying from 0.8 to 1.2 for cast iron and slightly smaller for mild steel.

Case I. Circular plates.

(*l*) *Uniform load, fixed at edges. Fig.* 17.

In this case the largest stresses are those in a radial direction at the circumference,

$$f_{\text{true}} = \frac{3}{4}\left(\frac{m^2-1}{m^2}\right)\frac{pr^2}{t^2}$$

or, for $m = 3$,

$$f_{\text{true}} = \frac{2\,pr^2}{3\,t^2}$$

$$t = \sqrt{\frac{2\,pr^2}{3\,f_{\text{true}}}}. \tag{40}$$

The stress at the center is one-half of the above.

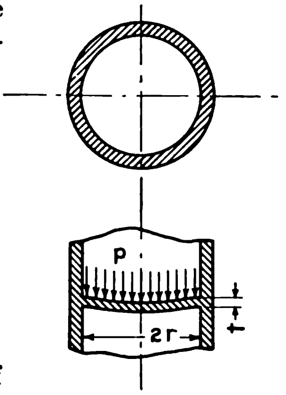


FIG. 17.

$$v = \frac{pr^4}{6\,Et^3}. \tag{41}$$

The maximum apparent stress is at the circumference.

$$f = \frac{3\,pr^2}{4\,t^2}. \qquad (42)$$

$$t = \sqrt{\frac{3\,pr^2}{4\,f}}. \qquad (43)$$

At the center

$$f = \frac{pr^2}{2\,t^2}. \qquad (44)$$

The general rules of the Board of Supervising Inspectors give, for unstayed wrought iron or steel flat heads not exceeding 20 inches in diameter, an equation for the allowable working pressure

$$p = \frac{512\,ct^2}{\pi r^2},$$

where c is 112 for plates $\frac{7}{16}$ inch and under in thickness, and 120 for plates more than $\frac{7}{16}$ inch thick.

Using $c = 112$, the above formula corresponds to a fibre stress of 15,200 pounds per square inch in the theoretical formula for a supported plate and 12,200 pounds per square inch for a plate fixed at the edges.

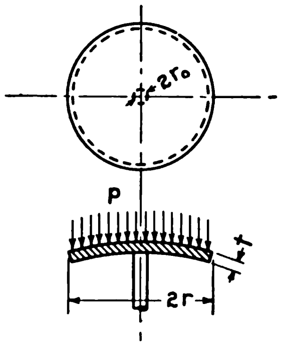


FIG. 18.

Case I. Circular plates.

(*o*) *Uniform load, free at edges, and supported at center by a circle of radius* r_0. *Fig.* 18.

$$f = \left(\frac{4}{3}\log_e\frac{r}{r_0} + \frac{1}{6}\right)\frac{pr^2}{t^2}, \qquad (45)$$

$$v = \frac{pr^4}{Et^3}. \qquad (46)$$

Case I. Circular plates.

(*x*) *Load P, distributed over a circle of radius* r_0, *supported at edges. Fig.* 19.

$$f_{\text{true}} = \frac{3}{2\,\pi}\left(\frac{m^2-1}{m^2}\right)\left(\log_e\frac{r}{r_0} + \frac{m}{m+1}\right)\frac{P}{t^2},$$

or, for $m = 3$,

$$\text{Max. } f_{\text{true}} = \frac{P}{\pi t^2}\left(\frac{4}{3}\log_e\frac{r}{r_0} + 1\right). \qquad (47)$$

To facilitate the use of the above equation the following values of f for the ratios $\frac{r}{r_0}$ shown are given:

$\frac{r}{r^0}$	10	20	30	40	50
f	$4.07\frac{P}{\pi t^2}$	$4.99\frac{P}{\pi t^2}$	$5.53\frac{P}{\pi t^2}$	$5.92\frac{P}{\pi t^2}$	$6.22\frac{P}{\pi t^2}$

For small values of $\frac{r_0}{r}$

$$v = \frac{5\,Pr^2}{3\,\pi Et^3}. \quad . \quad . \quad . \qquad (48)$$

The maximum apparent stress is

$$f = \frac{P}{\pi t^2}\left(\frac{3}{2} + 2\log_e\frac{r}{r_0}\right). \quad . \quad . \qquad (49)$$

With the above equation as r_0 approaches zero the stress approaches an infinite value. This is overcome by Bach, whose equation is of the form

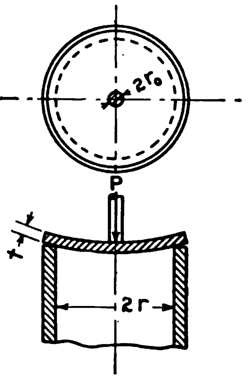


FIG. 19.

$$\text{Max.}\, f = \frac{3\,\mu P}{\pi t^2}\left(1 - \frac{2\,r_0}{3\,r}\right), \quad . \quad . \quad . \quad . \quad . \qquad (50)$$

or

$$t = \sqrt{\frac{3\,\mu}{\pi}\left(1 - \frac{2\,r_0}{3\,r}\right)\frac{P}{f}} \quad . \quad . \quad . \quad . \qquad (51)$$

For $r_0 = 0$ these equations become

$$f = \frac{3\,\mu P}{\pi t^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (52)$$

$$t = \sqrt{\frac{3\,\mu P}{\pi f}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (53)$$

In general μ, the experimental constant, may be taken 1.5, in which case

$$f = \frac{4.5\,P}{\pi t^2}\left(1 - \frac{2\,r_0}{3\,r}\right). \quad . \quad . \quad . \quad . \quad . \qquad (54)$$

This equation is to be used when r_0 is approximately $0.1\,r$, which agrees well with the theoretical formula, equations (47) and (54) being equal for a ratio of r_0 to r of 0.089. Bach states that for larger values of r_0, the coefficient should be decreased which is also in accord with formula (47).

Case I. Circular plates.

(*y*) *Load P, distributed over a circle of radius r_0, fixed at edges. Fig.* 20.

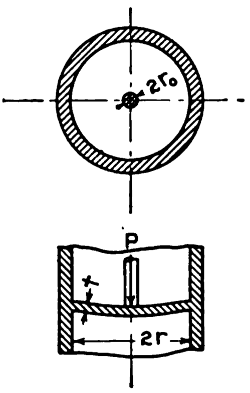


FIG. 20.

The stress at the center will be greater than that at the edges for all values of r_0 less than $\frac{1}{3}r$, which cases only are considered.

$$f = \frac{3P}{2\pi t^2}\left(\frac{m^2-1}{m^2}\right)\log_e \frac{r}{r_0}.$$

For $m = 3$,

$$\text{Max. } f_{\text{true}} = \frac{4P}{3\pi t^2}\log_e \frac{r}{r_0}. \quad . \quad (55)$$

For small values of $\frac{r_0}{r}$,

$$v = \frac{2Pr^2}{3\pi E t^3}. \quad . \quad . \quad . \quad (56)$$

To facilitate the use of equation (55) the following values of f corresponding to the ratios $\frac{r}{r_0}$ shown are given:

$\frac{r}{r_0}$	10	20	30	40	50
f	$3.07\frac{P}{\pi t^2}$	$3.99\frac{P}{\pi t^2}$	$4.53\frac{P}{\pi t^2}$	$4.92\frac{P}{\pi t^2}$	$5.22\frac{P}{\pi t^2}$

The stress is less in each case than the corresponding one with the plate supported at the edges, as is to be expected.

Case II. Elliptical plates.

(*k*) *Uniform load, supported at edges. Fig.* 21.

Let a = major axis.
b = minor axis.

Horizontal flexural stresses are greatest in a direction parallel to the minor axis, tests* of such plates developing cracks along the major axis.

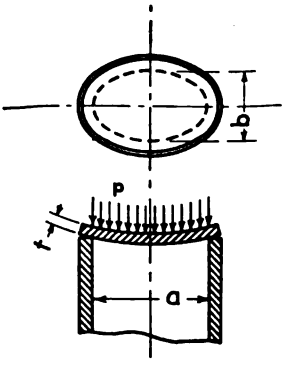


FIG. 21.

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* Bach, "Elastizität und Festigkeit," page 620, Figs. 3 and 4.

$$f=\frac{\mu}{2}\left(\frac{3.4+1.2\left(\frac{b}{a}\right)^2-0.6\left(\frac{b}{a}\right)^4}{3+2\left(\frac{b}{a}\right)^2+3\left(\frac{b}{a}\right)^4}\right)\frac{pb^2}{t^2},$$

or

$$f=\frac{\mu}{2}\left(\frac{a^2b^2}{a^2+b^2}\frac{p}{t^2}\right)\text{ very nearly. . . . . . }\quad(57)$$

Bach used a ratio of $\frac{a}{b}=\frac{3}{2}$ and found

$$\mu=\frac{9}{8}$$

hence

$$f=\frac{9\,a^2b^2p}{16\,t^2\,(a^2+b^2)}\text{ for cast iron . }\quad(58)$$

For wrought iron and steel use

$$f=\frac{a^2b^2p}{2\,t^2(a^2+b^2)}\text{ . . . . . . }\quad(59)$$

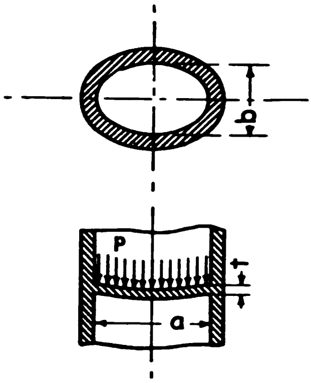

FIG. 22.

Case II. Elliptical plates.

(*l*) *Uniform load, fixed at edges. Fig. 22.*

Use equation (57) with constants such that

$$f=\frac{3\,a^2b^2p}{8t^2\,(a^2+b^2)}\text{ for cast iron . . . . }\quad(60)$$

and

$$f=\frac{3\,a^2b^2p}{9\,t^2(a^2+b^2)}\text{ for wrought iron and steel. }\quad(61)$$

Case II. Elliptical plates.

(*x*) *and* (*y*) *Load P, distributed over a circle of radius* r_0, *the plate having more or less fixedness at the edges. Figs. 23 and 24.*

$$f=\frac{8}{5\pi}\mu\left(\frac{8+4\left(\frac{b}{a}\right)^2+3\left(\frac{b}{a}\right)^4}{3+2\left(\frac{b}{a}\right)^2+3\left(\frac{b}{a}\right)^4}\right)\frac{bP}{at^2},\text{ . . . }\quad(62)$$

where μ varies from $\frac{3}{2}$ to $\frac{4}{3}$ depending upon the degree of fixedness at the circumference. The former quantity applies to the case of plates rigidly fixed at the edges, and the latter to those supported only. The radius r_0 must be small and the load essentially concentrated.

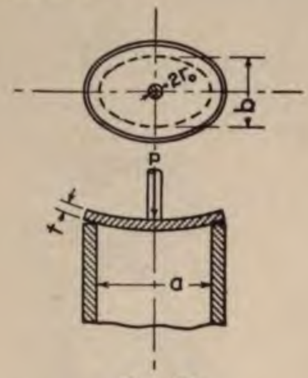

Fig. 23.

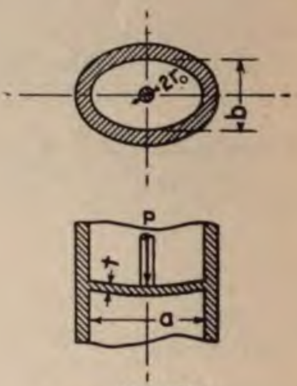

Fig. 24.

Case III. Rectangular plates.

(*k*) *and* (*l*) *Uniform load, the plate having more or less fixedness at the edges. Fig.* 25.

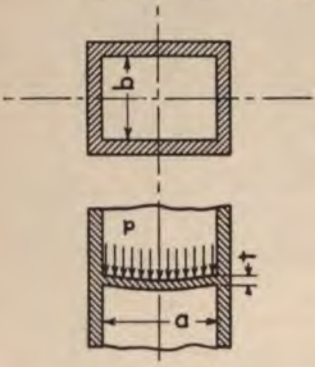

Fig. 25.

Relying upon the experimental work of Bach to determine the values of the constant in an equation of the form

$$f = \frac{\mu}{2}\left(\frac{a^2}{a^2+b^2}\right)\frac{b^2 p}{t^2}. \quad . \quad (63)$$

μ may be taken from $\frac{5}{8}$ to $\frac{3}{4}$, depending upon the degree of fixedness at the edges, the former value corresponding to entire flexibility and the latter to a good degree of rigidity.

Case III. Rectangular plates.

(*x*) *and* (*y*) *Load P, distributed over a circle of radius r_0, the plate having more or less fixedness at the edges. Fig.* 26.

$$f = \frac{3\mu}{2}\left(\frac{a\,b}{a^2+b^2}\right)\frac{P}{t^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

where $\mu = 1\frac{3}{4}$ to 2.

ere again the lesser value applies to cases of rigidity and the ter to those of flexibility.

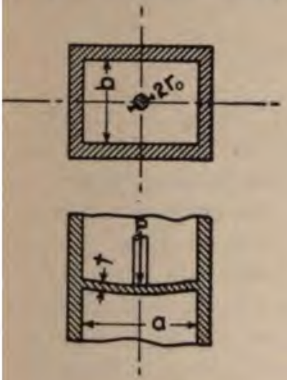

Fig. 26.

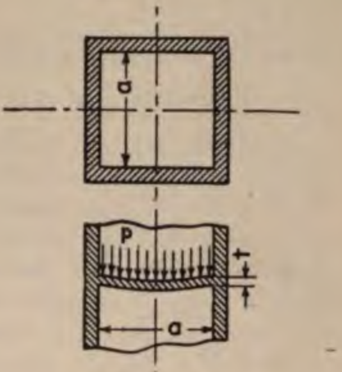

Fig. 27.

se IV. Square plates.

(*k*) *and* (*l*) *Uniform load, the plate having more or less fixedness at the edges. Fig.* 27.

om the equation given in *Case III* (*k*), for flexible edges,

$$f = \frac{9\,a^2p}{32\,t^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

imilarly from *Case III* (*l*), for fixed edges,

$$f = \frac{3\,a^2p}{16\,t^2}. \qquad (66)$$

IV. Square plates.

x) *and* (*y*) *Load P, distributed over a circle of radius* r_0, *the plate having more or less fixedness at the edges. Fig.* 28.

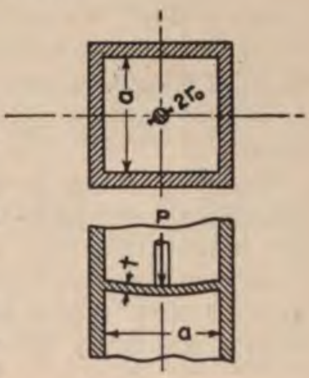

Fig. 28.

cases are special ones under ral discussion of rectangular en above. Applying formula dges merely supported,

$$f = 1.50\left(\frac{P}{t^2}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

and for fixed edges,

$$f = 1.31\left(\frac{P}{t^2}\right). \quad \ldots\ldots\ldots \quad (68)$$

Case V. Continuous plates.

(*l*) *Uniform load, stayed at a system of points. Fig.* 29.

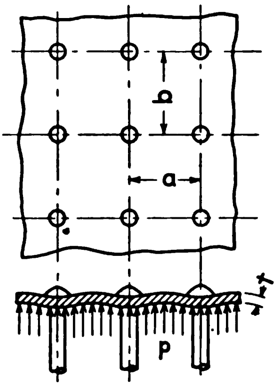

Fig. 29.

The problem of a continuous flat plate supported at a series of points, usually pitched on rectangles or squares, is an extremely important one. A theoretical discussion of this case is given by Grashof in which it is assumed that any strip connecting two adjacent rows of rivets is in the condition of a beam fixed at the ends. The stress corresponding with the maximum strain is found to be indeterminate when a is not equal to b, Fig. 29. When a is equal to b

$$f_{\text{true}} = \left(1 - \frac{1}{m^2}\right)\frac{a^2p}{4\,t^2}.$$

Taking $m = 3$.

$$\text{Max. } f_{\text{true}} = \frac{2\,a^2p}{9\,t^2},$$

$$a = \sqrt{\frac{9ft^2}{2p}}, \quad \ldots\ldots\ldots \quad (69)$$

which is intermediate between the values given for square plates as determined from experiment, i.e., Case IV (*k*) and (*l*) given above.

Experimental data on stayed flat plates, covering 59 tests in 10 different sets of experiments, have been carefully summarized by Mr. C. E. Stromeyer *. He concluded that "no matter what the mode of staying may be, the pressure at which the first permanent set takes place in plate of 60,000 pounds per square inch tenacity is found by the equation"

$$p = \frac{667\,t^2}{D^2}.$$

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* The Engineer, March 13, 1914. Engineering, Feb. 6, 1914.

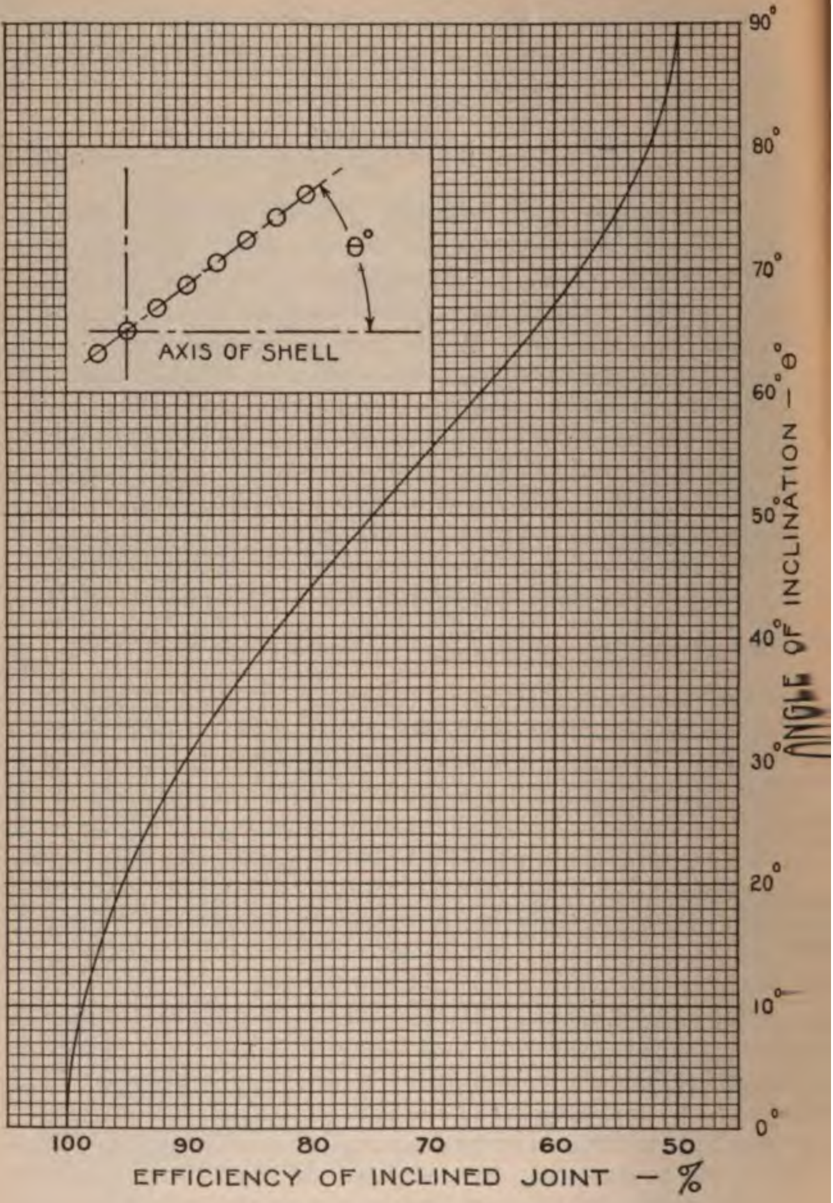
D is the diameter of the largest circle which can be inscribed inside of the stays, and t is the thickness of plate in sixteenths of an inch. With t expressed in inches

$$p = \frac{170{,}700\,t^2}{D^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

A comparison of this formula with the theoretical formula, assuming $D^2 = 2\,a^2$, will show a value of $f = 19{,}000$ lbs. per sq. in., which is low for the elastic limit of wrought iron or steel plate. This is partly due to the assumption $D^2 = 2\,a^2$ as D^2 is always less than $2\,a^2$. As an example, assume 1 inch rivets, spaced 6 inches on centers. $a^2 = 36$, $D^2 = 56.03$, hence $D^2 = 1.556\,a^2$ and the equations show f to have a value of 24,400 lbs. per sq. in., which is a conservative value for the elastic limit. Where the ratio of diameter of stay to distance on centers is less than one-quarter, equation (70) will show a small factor of safety on an elastic limit of 28,000 lbs. per sq. in.

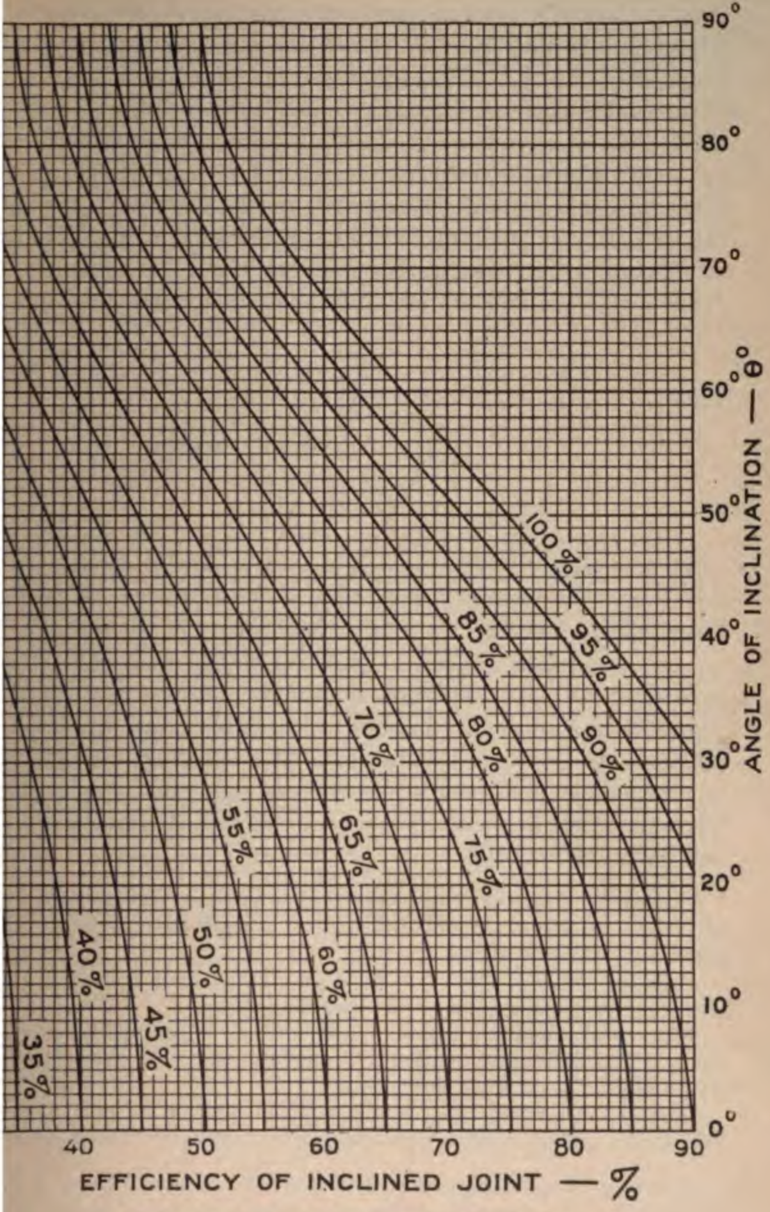
The Board of Supervising Inspectors' rule for stayed flat plates ("all stayed surfaces formed to a curve the radius of which is over 21 inches") is $p = \frac{256\,Ct^2}{a^2}$, where a is the greatest pitch of stays, and C is a constant varying from 112 to 200, the smaller value being for screw stays with riveted heads. Comparing this with the theoretical formula (69) the smaller constant, 112, is found to correspond to a working fibre stress of 6370 lbs. per sq. in., and the larger one, 200, to a fibre stress of 11,400 lbs. per sq. in. The form of the equation in the Massachusetts Boiler Rules is such that it does not admit of a ready comparison with formula (69). It may be said, however, that for ordinary pressures and thickness of plates the results agree very closely for working fibre stresses of from 5000 to 5500 lbs. per sq. in.

22. Helical Seams. — It has been shown, page 58, that in the case of a thin cylindrical shell subjected to internal pressure the apparent stress on a longitudinal cross section is twice that on a transverse cross section. These planes being principal planes (see page 51) one stress is a maximum, the other a minimum. Therefore, the stress on any inclined plane will be less than that on a longitudinal section, hence, for a given pressure, an inclined joint need not have as high an efficiency as would be required were the joint arranged longitudinally. This has given rise to the produc-



HELICAL JOINTS
OF
MAXIMUM EFFICIENCY

Fig. 30A.



EQUIVALENT EFFICIENCY
OF
HELICAL JOINTS

Fig. 30B.

tion of helically riveted joints on cylindrical shells, especially where the diameter is small and the length comparatively large as in the case of pipes. For large diameters and short lengths, as in steam boilers, inclined joints have not come into favor, due to the necessary loss of plate. While the resultant stress is not normal to the center line of the joint, there being a component of shear on this plane, it is customary to use the resultant stress in calculating the efficiency of the joint. From equation (1) at the opening of this chapter, the stress on any inclined joint may be found.

Let θ = angle inclined joint makes with longitudinal joint.

$$p_x = 2\,p_y.$$

Then
$$\begin{aligned} p_r &= \left[p_x^2\cos^2\theta + \left(\frac{p_x}{2}\right)^2\sin^2\theta\right]^{\frac{1}{2}} \\ &= \tfrac{1}{2}\left[4\,p_x^2\,(1-\sin^2\theta) + p_x^2\sin^2\theta\right]^{\frac{1}{2}} \\ &= \frac{p_x}{2}\,(4 - 3\sin^2\theta)^{\frac{1}{2}}. \qquad \text{(71)} \end{aligned}$$

To facilitate the solution of problems involving the use of equation (71), two plots, Figs. 30A and 30B, have been prepared. From the former may be read the efficiency of an inclined joint necessary to make the seam as strong as the solid plate, that is, to correspond to a longitudinal joint of 100 per cent efficiency. Fig. 30B is similar to 30A, but is somewhat more general in its application. From it may be found the corresponding efficiencies of an inclined joint and a longitudinal joint, for equal strength, when the angle between the joints is known. Concrete examples will best illustrate the use of the plots.

Problem 1. Find the required efficiency of a helical joint making an angle of 75° with the axis of the shell if the joint is to make the cylinder the equivalent of one without a seam.

From the plot, Fig. 30A, it is found that for 75°, an efficiency of 54.8 per cent will give a helical seam of maximum efficiency.

Problem 2. In an air tank it is desired to use a joint of 54 per cent efficiency. The angle of inclination of the joint is 26°. To what efficiency of longitudinal joint does this correspond?

Interpolating the plot, Fig. 30B, it is found that for 26° and 54 per cent efficiency of inclined joint, the efficiency of longitudinal joint is 58.5 per cent.

Helically riveted pipe is on the market * in sizes from 3 inches to 42 inches diameter, and thicknesses varying from 0.0375 inch to

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* American Spiral Pipe Works.

0.1406 inch for standard pipe. Three widths of strips are used, viz., 9, 12 and $15\frac{1}{4}$ inches, with laps varying from $1\frac{1}{8}$ to $1\frac{3}{4}$ inches. With these dimensions, in all cases except the 3 and 4 inch pipes, the full strength of the pipe may be obtained with a joint of less than 60 per cent efficiency, which is readily obtained with a single-riveted lap joint. This increase of effectiveness of the joint when inclined may be taken advantage of in patching cylindrical shells, the joints being inclined rather than longitudinal.

CHAPTER III.

FASTENINGS BY RIVETED JOINTS.

23. Rivets.—The rivet constitutes the simplest form of fastening. It consists essentially of a permanent bolt, head, nut and body forming one piece. When once set in place it cannot be removed except by chipping off the head. Bolts and screws are usually arranged to hold their loads by axial tension. Owing to the flexibility in rivet heads, such fastenings are not considered wholly reliable when subjected to tension. Hence, when possible, rivets are set to bear their loads by shear. In some cases, such as the staying of the walls of certain pressure vessels, a small amount of flexibility is desirable to accommodate expansion and contraction. The rivet under such conditions is well adapted to hold a tensile load.

Rivets are hot-pressed to shape from round bar stock. The shank is sometimes slightly tapered near its end to facilitate its entrance into the hole. One head and the shank are finished at the time of manufacture though the shape of the head may be altered later, due to the method of driving the rivet. For ease in placing rivets in position, their shanks are made one-sixteenth inch less in diameter than that of the holes they are to fill. When well set, however, they are assumed to fill the holes entirely. Therefore the actual diameter used in calculation, and that specified on design drawings, is the driven or hole diameter.

24. Driving Rivets. — Rivets may be set when either hot or cold, depending upon the character of the work and the rivet material. Small and unimportant rivets may be set when cold provided they are made of soft, ductile metal which will not be injured by rough treatment. Large and important work, however, is practically always finished by shrinking hot rivets to place under intense pressure. When setting field-driven rivets and working on repairs, a portable hand forge, using screened coke or coal, is generally employed to heat the rivets to a bright red. For permanent use in the shop a petroleum or gas oven is best adapted. Great care must be exercised to avoid over-heating and burning the rivets. If left in the fire to "soak" at full heat for long periods, the character of the rivet material is changed and its strength greatly reduced.

In general there are three methods of finishing the work:

(*a*) *Hand riveting;*
(*b*) *Machine riveting;*
(*c*) *Pneumatic riveting.*

The importance and accessibility of the work together with the appliances at hand generally determine which method shall be used.

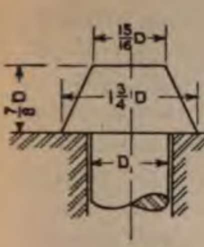
(*a*) *Hand Riveting.* — The hot rivet, after insertion in the hole, is held in place by a boiler-maker's anvil securely braced against some other part of the structure. Workmen striking alternating blows with hand hammers beat the head to approximately the correct form. A cupping tool, or "snap," containing a cavity of the exact contour desired for the driven rivet head, is placed over the partially finished work and given a few sharp blows with a sledge hammer. This gives the head the final shape and finish. There is always more or less lack of uniformity in hand-riveted work owing to differences in the heat and in the time spent in driving successive rivets. The skill of the workman has also much to do with the strength of such seams. With a faithful and skilful performance of the task the results, however, are very good. Owing to the increasing expense of hand-driven riveting, this method is seldom used at present. In rapid repair work and in some isolated localities where other methods are not possible, the hand-driven rivet is occasionally employed. Naturally the impact of the hammers and cupping tool tends to injure thin shells, and care must be taken that the anvil or "holder-on" is well braced against the rivet while it is being driven.

(*b*) *Machine Riveting.* — For machine riveting the inserted rivet is brought while red hot into line with the plunger of a steam or hydraulic press. Dies, carried by the plunger, are capable of exerting a pressure varying, with the size of the rivet, from twenty-five to one hundred and fifty tons. The rivet heads are thus accurately forced to form, the pressure being held upon them for a brief interval while cooling. A jet of water is arranged to play upon the rivet as soon as set and thus hasten its contraction. After having cooled to blackness the pressure is removed and consequently the rivet grips the plate with great intensity. Machine-driven rivets, when set in an accurate manner, show great evenness and uniformity of strength.

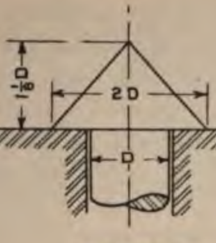
(*c*) *Pneumatic Riveting.* — A pneumatic riveting hammer consists of a vibrating air piston carrying the rivet die or "snap." The hot rivet, inserted in its hole, is held in place by a pneumatic anvil or holder-on. The latter consists of a heavy air piston and cylinder connected to pneumatic pressure. When braced against some solid portion of the work and subjected to air pressure this anvil holds the hot rivet securely in place. The pneumatic hammer is then brought to bear upon the protruding rivet shank and beats the head to shape before cooling. Many otherwise inaccessible places can be reached by the pneumatic riveter, and where the jar of the piston is not injurious to the work, excellent results are obtained.

Rivets set while hot grip the plates with great force due to their contraction upon cooling. Very long rivets, especially when formed of certain alloys, may contract to such a degree as to overstrain themselves upon cooling. Care must be taken to heat such rivets to dull redness only before driving. Sometimes the driven end only is heated. To a reasonable degree rivet contraction is desirable in order to keep the seam from leaking and to produce friction between the plates forming the joint.

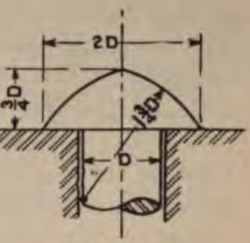
25. Rivet Heads. — While there is no exact agreement among rivet manufacturers or users as to the standard shape of rivet heads, Figs. 31 to 40 inclusive show the usual proportions in terms of the actual or undriven diameter. The particular form of the rivet head for a given case is determined by the purpose for which the joint is intended. When the riveting is designed merely to resist rupture, as for instance in the members of a roof truss, heads of smaller size and containing less material can safely be used. On the other hand when the seam must not only sustain severe stresses but be staunch enough to confine fluid pressure as well, heads of greater stiffness and amplitude must be used. Figs. 31, 32, 33 and 34 show the proportions of heads generally used in pressure work, as their width and depth would indicate. The cone head, Fig. 31, is the usual one pressed upon commercial rivets as they come from the manufacturers. In hand-riveted work this head appears on the under side of the seam. Fig. 32 illustrates the usual form of hand-driven rivet head. The straight



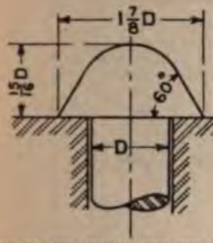
CONE
FIG. 31.



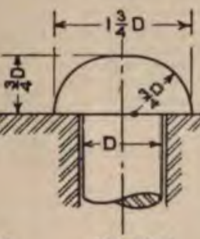
STEEPLE
FIG. 32.



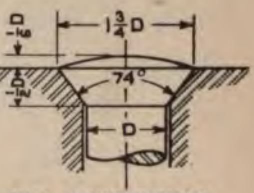
PRESSURE
FIG. 33.



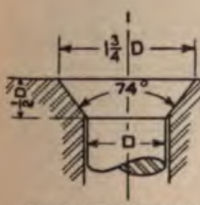
INTERNATIONAL ENGINEERING CO.
FIG. 34.



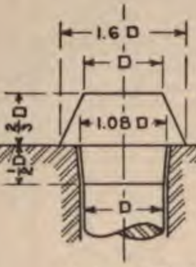
BUTTON
FIG. 35.



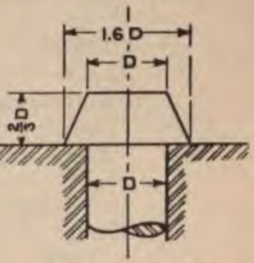
FULL COUNTERSUNK
FIG. 36.



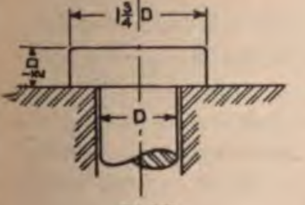
FLAT COUNTERSUNK
FIG. 37.



SWELL NECK
FIG. 38.



STRAIGHT NECK
FIG. 39.



FLAT
FIG. 40.

STANDARD PROPORTIONS
FOR
RIVET HEADS

slope of its sides can be accurately fashioned with a hand hammer. The thin edges render it too flexible for heavy pressures and the sharp point is liable to burn off when used in externally-fired boilers. The type most widely used in machine riveting is shown in Fig. 33. Its large diameter and staunch sides commend it for pressure use. A special head very deep and stiff, used by the International Engineering Company in their boiler shop, is illustrated by Fig. 34. The usual form of structural head appears in Fig. 35. Often the room for the head is restricted due to other parts of the machine or structure. In such cases the full- or flat-countersunk heads of Figs. 36 and 37 have to be used. When rivets are subjected to excessive vibration, tending to crack the shanks just underneath the heads, a conical enlargement is added where the shank and head join as shown in Fig. 38. Sometimes a rivet is designed to fit accurately a drilled hole all the way through and to be riveted into a collar or washer at the other

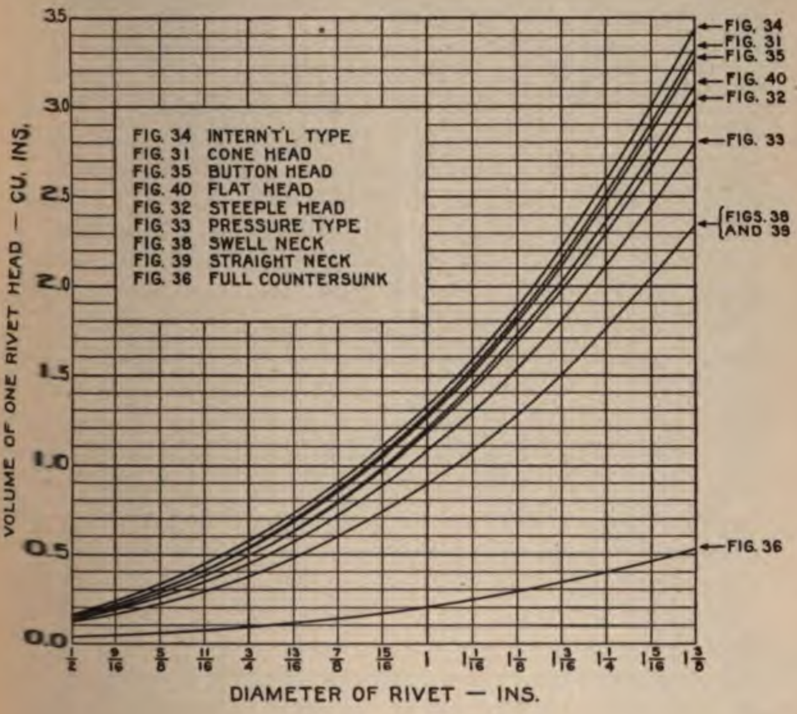
TABLE I.

PROPORTIONS OF STANDARD RIVET HEADS.

Figure.	Type.	Volume of one head in terms of actual or undriven shank diameter. Cu. ins.	Length of rivet shank necessary to join plates of total thickness T. Ins.
31	Cone...........	$1.279\ D^3$	$T+\frac{0.125\ T}{D}+1.63\ D$
32	Steeple.........	$1.178\ D^3$	$T+\frac{0.125\ T}{D}+1.50\ D$
33	Pressure........	$1.080\ D^3$	$T+\frac{0.125\ T}{D}+1.38\ D$
34	Inter. Eng. Co..	$1.323\ D^3$	$T+\frac{0.125\ T}{D}+1.69\ D$
35	Button.........	$1.268\ D^3$	$T+\frac{0.125\ T}{D}+1.61\ D$
36	Full countersunk	$0.203\ D^3$	$T+\frac{0.125\ T}{D}+0.73\ D+0.15$
37	Flat countersunk	....	$T+\frac{0.125\ T}{D}+0.47\ D+0.15$
38	Swell neck......	$0.901\ D^3$	$T+\frac{0.125\ T}{D}+1.15\ D$
39	Straight neck...	$0.901\ D^3$	$T+1.15\ D$
40	Flat............	$1.202\ D^3$	$T+\frac{0.125\ T}{D}+1.53\ D$

end. In such cases the straight shanked rivet of Fig. 39 would be used, no clearance being allowed between the rivet and hole. The flat head of Fig. 40 is used in close quarters where there is no external heat to burn off the sharp corners.

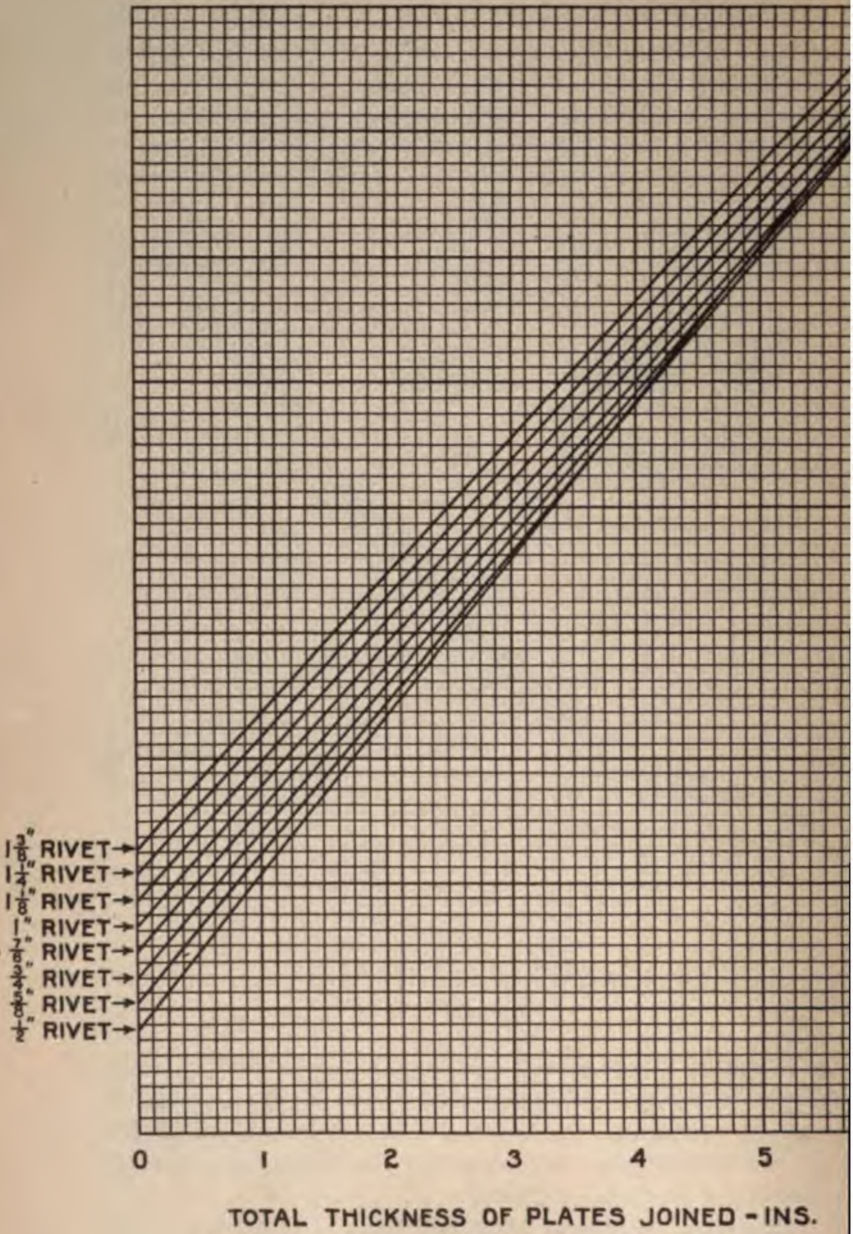
It is necessary to be familiar with the general proportions of rivet heads in order to judge accurately of the possibility of setting them in places where room is a minimum. Allowance must usually be made for heads approximately twice the diameter of the driven rivet shank. An occasional rivet head may be gouged away with a cold chisel to permit the assembling of successive portions of the machine or structure.



VOLUME OF RIVET HEADS

Fig. 41.

In calculating the weight of riveted work it is sometimes advisable to include the weight of the rivet heads. Table I gives



ALLOWANCE FOR RIVET HEADS

Fig. 42.

the volume of the various rivet heads in terms of the actual or undriven shank diameter. By considering the holes in the plate to be completely filled by the rivet shanks after driving, and adding the extra volume contained in the heads, a very close approximation to the correct weight may be reached. The volume given in the table corresponds in every case to that which extends outside the surface of the plate and does not therefore include the tapering portion of a countersunk head. The curves of Fig. 41 give the plotted values of rivet head volumes in cubic inches for various diameters of shank and for the several forms shown in the previous illustration.

In ordering rivets for use in various thicknesses of seam, care must be taken that the shank protrudes far enough beyond the surface of the plate before riveting to give sufficient stock for forming the head. In the last column of Table I is given the approximate length of shank necessary for the above purpose. This takes into account the volume necessary to fill the clearance, $\frac{1}{32}$ in., around the rivet as well as to form the head, where the total thickness of the plates joined is represented by T. Since these expressions are somewhat cumbersome to apply, the plot, Fig. 42, has been worked out to represent the length of rivet shank necessary in riveting through a total thickness of plate T, the assumption being that a full type of head is to be used corresponding to a volume of $1.3\,D^3$. If extra fullness is desired in the head, the next eighth or quarter inch above that given by the plot should be selected.

26. Holes. — The holes for receiving rivets are generally punched cold. Since but one thickness of plate can be punched at a time the holes in the two or more thicknesses to be riveted together rarely come in correct register. The drift pin has often to be called into use in such work to get the rivets into place prior to riveting. A still greater objection to this method is the lessening of the tensile strength of the metal around the holes, due to injury from punching. In thick plates especially, the pressure of the punch, combined with the difference in diameter of punch and die, causes a lateral flow of metal resulting in jagged-edged holes. The burr left by the punch greatly reduces the holding power of the rivets also. While the use of a helical punch producing a progressive shear around the hole lessens the injury somewhat, extensive tests have shown that the loss in tensile strength between holes due to punching is about 12 per cent. For the above reasons cold-punched holes in their natural condition are now seldom used for boiler work. In the best classes of marine work it is required by law that all rivet holes shall be drilled from the solid metal.

27. Preparing Rivet Holes. — The injury done to the plate by punching holes may be partially remedied in two ways:

(a) *Annealing;*
(b) *Reaming the holes.*

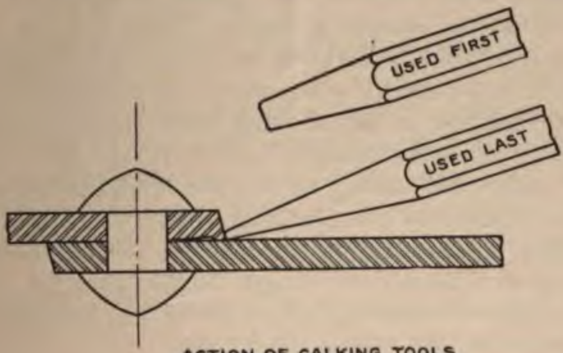
A thorough annealing after punching will restore in large measure the plate strength. The rough holes, however, are still severe in their action upon the rivets. It is generally impracticable to anneal portions of large sheets merely for the sake of the rivet holes. A more effective method is to drill or ream out the punched holes to size, thus removing the rough edges as well as the adjacent distorted plate. It is perfectly possible to ream several thicknesses of plate at once and thus secure a correct alignment of all the holes. After punching and bending the plates to shape, the tank or boiler may be assembled and all the rivet holes reamed in place. In the case of plates $\frac{1}{4}$ in. and less in thickness, the punched hole is made $\frac{1}{8}$ in. less in diameter than the hole desired after reaming. For $\frac{5}{16}$ in. plates and above, the punch must be $\frac{1}{4}$ in. less in diameter than the finished hole, under Massachusetts Boiler Rules. When punched holes are so treated the strength of the metal between them is unimpaired. In very careful work the plates are taken apart after reaming and the burrs around the holes upon both surfaces removed by a special countersink.

28. Classes of Riveted Work. — Riveted work may be divided into three broad classes according to its use:

(a) *Structural work;*
(b) *Boiler and tank work;*
(c) *Hull riveting.*

In structural work the holding power of the rivets, generally in shear but occasionally in tension, is the prime essential. In boiler work, besides the holding power of rivets as fastenings, the joints must embody a good degree of staunchness in order to be proof against leakage. In hull work the question of leakage presents but little difficulty, due to the low fluid pressures under consideration, while strength and durability are the chief requisites.

29. Calking. — Riveted work as it comes from the machine is never proof against leakage, but must be calked tight. For this reason riveted plates are always given a certain overlap. The distance of the rivet center back from the edge of the plate is governed by conditions of staunchness quite as much as of strength. There must be metal enough ahead of the rivets in all cases to prevent them from rupturing the lap. At the same time the lap must be a proper one to facilitate calking to tightness. The action of calking is represented by Fig. 43. The flat-ended tool shown above, actuated either by hand or pneumatic hammer, gives the lap edge a preliminary beating down and leaves a thin sharp edge. A second tool having a narrower end is then substituted for the previous one and this edge is driven back beneath the upper plate. In driving back the edge to tightness the upper plate is severely sprung up and reacts against the lower one after the manner of a deflected cantilever, with sufficient intensity to keep the fluid pressure from passing the calked edge. Very frequently the careless use of calking tools with sharp corners cuts into the lower plate to a degree sufficient to cause injury. Great care should be taken that the operation of the calking tool is confined to its legitimate place at the edge of the upper sheet.



ACTION OF CALKING TOOLS

FIG. 43.

The passage of the pneumatic calking tool over a seam should be steady and continuous. To secure the best results the distance between rivets on calked edges should be kept uniform, all the pitches of plate being calked therefore with equal tightness. It is very difficult to calk around sharp rectangular corners, the action of the tool along one side of the angle serving to open t other side. Such corners, therefore, should be chipped to a radi of three-fourths of an inch or more. The calking tool if h normal to this curve when passing around the corner will secu uniform tightness. When the outside cover plates of butt-joi are thinned and widened at their ends to facilitate their inserti under external courses at the ring seams of cylindrical shells, t irregular corners resulting should be chipped to a concave of ab one inch radius. The passage of the calking tool along the e of the outside cover plate and around the concave will me without interruption into the ring seam and produce tight wo

30. Lap Limits. — To be proof against leakage the lap acti like a deflected cantilever must react against the lower plate, Fig. with an intensity of pressure greater than that of the fluid with Tests have shown that while there are areas under the rivet he where the fluid pressure does not penetrate, in general the calk edge itself, especially in new work, is the final barrier to leaka It is perfectly evident, therefore, that short laps and rivets cl together render the calking easy, while large pitches and lo laps, due to continuous springing back, render the process diffic On the other hand small laps are liable to fail by bursting o while large ones remove all danger in this direction. A sa factory method of designing laps, when considered from both the above standpoints, will be given later.

31. Test Pressure. — Boilers and closed tanks are ordina calked to tightness under cold-water test pressure equal to and one-half times the working pressure, as was noted under Massachusetts State Rules in Chapter I. Usually the outs seams only are calked. In marine boilers and other very ca fully riveted vessels, a preliminary calking is given to the ins laps. Absolute tightness is finally secured, however, by calk on the outside.

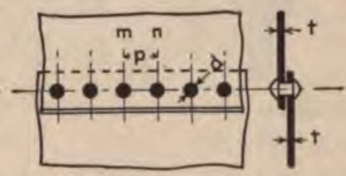
32. Friction between Plates. — The question of friction tween plates is regarded with varying degrees of importan American designers for the most part recognize its presence desirability in enhancing the holding power of riveted joints doubt its value as a basis for calculation. European writers, ho ever, have held this feature in much higher esteem and have go so far as to base a theory of joint design upon the intensity the friction between plates. Inasmuch as it is known that join slip or "take up" a little, while in use, it seems hardly possible that friction between plates can be relied upon as a holding power in joints. When plates are new and clean there is considerable slip discernible in the testing machine long before the working load of the joint is reached. An extended series of tests upon the holding power of riveted joints due to friction was made at the U. S. Arsenal at Watertown, Mass., in 1882. These tests show a wide variation in results depending upon the condition of the plate surfaces, but in all cases indicate considerable slippage at loads constituting but a fraction of the ultimate strength.

33. Riveted Joint Failures. — Riveted joints may give way in either of four different ways:

(*a*) *Tearing of the plate;*
(*b*) *Failure of the rivets;*
(*c*) *Tearing of plate and rivet failure combined;*
(*d*) *Lap rupture.*

As will be explained later the design of the lap will be carried out in such a way as to remove at the outset all possibility of the failure (*d*). Hence the first three failures are the only ones to be considered in the design of joints. Before making an application to any specific joint, the above failures will be discussed in the abstract.

(*a*) *Tearing of the Plate.* — The most usual method of failure is by tensile rupture between rivet holes accompanied by a distortion of the joint due to bending. In some forms of joints the fluid confined within may so corrode the plate as to reduce its thickness and thereby render it still more liable to tear. In other joints an inside cover plate protects the main plate between rivet holes from corrosion. Such a cover plate is always made much thicker than is necessary for the strength of the joint alone, in order to provide plenty of metal to resist corrosion. Fig. 44 shows the simplest form of joint, called a single-riveted lap joint. In discussing the resistance to failure for any joint an elementary portion called a "repeating section" must first be chosen. Thus, in Fig. 44, if the portion *mn* be repeated indefinitely, any length of joint may be obtained and whatever is true of the por-



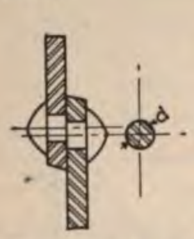
SINGLE RIVETED LAP JOINT

Fig. 44.

tion *mn* will be true of the whole joint. With the significance of the letters shown in the figure and f_t, the tensile strength of the plate, the resistance to tearing on the line between rivet holes will be, for the section *mn*,

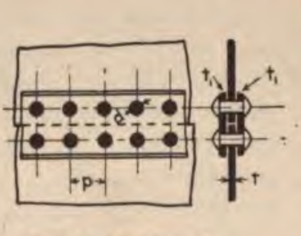
$$R_t = (p - d)\, t f_t.$$

If the distance between rivets is small the actual fibre stress per square inch at failure will considerably exceed the usual tensile strength f_t, on account of the necked or grooved specimen to which the metal between rivets is equivalent. The reason for this increase in tensile strength on the rivet line is due to the reinforcing action which the adjacent portions of plate exert upon the reduced sections between holes. In some cases this enhancement of value may reach from 10 to 25 per cent. While it is certain that this very desirable effect is present in all joints to a more or less degree, the advantage thus gained is rather to be relied upon to compensate for the injury due to punching than to be considered as so much net increase in strength. It is not customary, therefore, to make any allowance for grooved specimen effect in riveted joints. Complicated joints with large pitches do not participate very greatly in this gain in strength, while in simple unimportant ones, with rivets close together, the reverse is true. Therefore, whatever gain there is, shows itself where least needed.



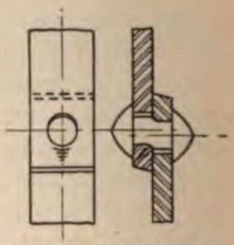
FAILURE BY SINGLE SHEAR

Fig. 45.



DOUBLE SHEAR JOINT

Fig. 46.



FAILURE BY CRUSHING

Fig. 47.

(*b*) *Rivet Failures.* — The second method of failure in riveted joints consists in the destruction of the rivets. If the rivets are small in diameter and of soft material, while the boiler plate is thick and hard in texture, stress on the joint of Fig. 45 may result in shearing the rivet at the line where the plates touch. The two portions of the rivet so sheared are separated sharply and smoothly.

Neither the rivet shank nor the surrounding plate suffers much distortion by the operation. The load at which this failure would take place is represented by the expression

$$R_s = \frac{\pi d^2}{4} f_s$$

where f_s equals the shearing strength of the rivet material.

Sometimes rivets are so located in a joint as to shear upon two sections at once. Fig. 46 represents a joint where the main plates abut against one another, and two cover plates or "butt straps" provide the connection. In case of the rivets shearing, two sections of each rivet would be ruptured simultaneously. The theoretical resistance to this failure is

$$R_s = \frac{\pi d^2}{2} f_s.$$

Another quite different manner of rivet failure is produced by excessive bearing pressure. Imagine the rivet large in diameter and the plate comparatively thin. An increasing load upon the joint will cause a distortion of the rivet hole, as shown in Fig. 47, fatally impairing the holding power of the rivet. The crushing of the rivet and plate is generally mutual, namely, both plate and rivet are severely distorted by the action. This failure is generally made evident by the slippage of one plate upon the other, rendering the joint unfit to hold its load or to confine fluid pressure. Even though the ultimate destruction of the joint may result in shearing the rivets, the fundamental failure would have to be attributed to excessive bearing pressure between rivet and plate. It is difficult to analyze the distribution of load between a round rivet shank and the plate ahead of it. For this reason bearing pressures are calculated per unit of projected area. The projected area of the plate ahead of the rivet equals *dt*, and this, multiplied by the mutual crushing strength f_c of the rivet and plate as determined from tests under similar conditions, gives the total resistance. The resistance to failure of this kind may then be expressed as

$$R_c = dtf_c.$$

Many tests have been made to determine the relation between the values of f_s for single and double shearing under practical conditions. Single-shear tests have generally been made upon joints similar to Fig. 44. As will be pointed out later such joints tend to straighten out under a load, with the result that considerable tension comes on the rivets. In double-shear tests, upon joints similar to Fig. 46, the rivets are suitably located to receive their loads without distortion. The friction between plates also tends to increase the load necessary to shear the rivets. It would seem, therefore, that the strength of a rivet in double shear ought to be at least twice as much as in single. In many tests along this line the rivets were so large and the consequent bearing pressure so great, that failure should have been attributed to crushing rather than to double shearing. Even though the rivets were finally dragged apart at a low figure in double shear, the real failure was due to excessive bearing pressure. Before crushing was recognized as a rivet failure, various Boards of Inspection and Insurance arbitrarily fixed the value of double shear as less than twice single shear in order to keep the bearing pressure upon the rivets sufficiently low. Such is the case at present in the Massachusetts Boiler Rules. It appears much the more logical way to assume that double shear is twice single shear and then to give careful attention to the bearing pressure as a separate item.

A series of tests made upon the Government Testing Machine at the U. S. Arsenal, Watertown, Mass., throws some light upon the question of the relative strength of rivets in single and double shear. The tests are reported in "Tests of Metals," 1882 and 1886. The average shearing strength of the rivets in nineteen single-shear joints was 39,480 lbs. per sq. in. Under identical conditions, thirty-two double-shear joints showed an average rivet strength of 39,040 lbs. per sq. in. The joints were entirely comparable in every respect and the same grade of soft rivet iron was used throughout. From these tests it would appear that the shearing strength per square inch may be taken the same in single and double shear.

(c) *Combined Rivet and Plate Failure.* — In most complicated joints with several rows of rivets, there is liability to failure consisting of tearing the plate upon one line accompanied by the destruction of the rivets upon another. The mathematical form of the resistance to this failure varies so widely with the type of the joint that no general expression for it can be given. The expression consists of the tearing strength of the plate added to the rivet failures necessary to permit the tearing to take place.

(*d*) *Lap Failure.* — A fourth method of failure for riveted joints is by lap rupture. With very thin plate a rivet may be imagined to plow its way to the lap edge, as shown at *b* in Fig. 48, removing the plate in front of it by shear. As a matter of fact with practical sizes this rarely happens. The way in which lap rupture ordinarily occurs is represented by *a*, Fig. 48. The metal ahead of each rivet is loaded by the tension in the joint much after the manner of a minute beam. The failure of the lap shows a typical beam rupture by breaking on the lower or tension side near the center.

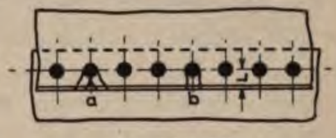


Fig. 48.

Fig. 49 illustrates the appearance of a riveted-joint specimen after the laps had been ruptured. This joint is one of a series tested in 1898 by Messrs. Wilder & Wesson in the laboratories of the Massachusetts Institute of Technology. The appearance of the ruptured laps is very significant. The edge opened under the rivets in a manner entirely similar to the rupture of a beam and, if the test could have been stopped at that point, the method of failure would have been still more evident. Furthermore, the load at rupture corresponded very closely with that derived from calculating the laps as beams.

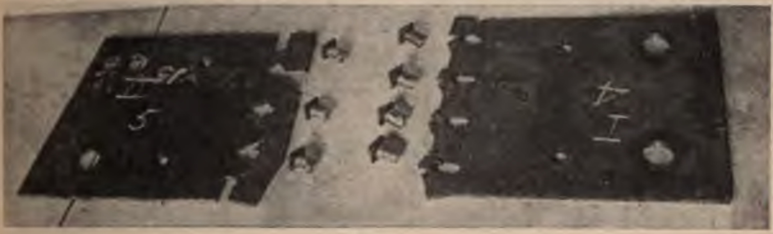


Fig. 49.

A method of lap design will later be explained, having the beam theory for its basis. The amount of lap is largely under the control of the designer and if the former is properly determined, this method of failure may be entirely eliminated from the joint, as was previously stated. Frequently the lap is arbitrarily assumed to be one and one-half times the driven rivet diameter.

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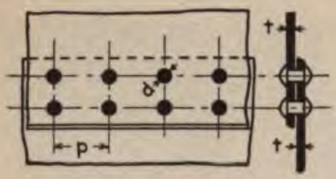
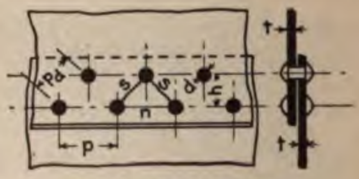
34. Arrangements of Rivets. — When several rows of rivets are used in lap joints the rivets may be arranged in either of two ways:

(a) *Chain riveting;*
(b) *Staggered riveting.*

If one rivet is placed directly behind the other, as shown in Fig. 50, the joint is designated as chain riveted. This method is generally used in structural work. When, however, fluid pressure is to be confined, as well as great strength secured, the rivets should be disposed as shown in Fig. 51. This is called staggered riveting, and greatly enhances the value of the joint as a means of holding fluid pressure. In order to make the calculation of the joint as definite as possible, the tearing of the plate should take place along a horizontal rivet line such as n rather than upon the oblique lines s and s. It has been well established by experiment that if 30 per cent more tearing area is placed on the oblique lines s and s than on n, the rupture will be confined to the line n. Therefore the approximate relation



CHAIN RIVETING
FIG. 50.



STAGGERED RIVETING
FIG. 51.

$$\frac{s+s}{n}=\frac{4}{3}$$

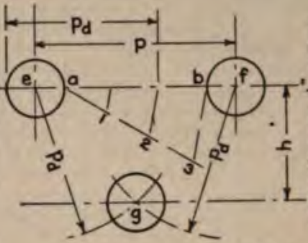
should be observed in arranging the rivets in staggered rows. Taking the significance of the letters suggested in Fig. 52, where p_d equals the slanting or diagonal pitch, and observing the above relation,

$$\begin{aligned} p_d &= \tfrac{2}{3}(p-d)+d \\ &= \frac{2p+d}{3}. \end{aligned}$$

Then h, the distance between rows, will be

$$h=\sqrt{\left(\frac{2p+d}{3}\right)^2-\left(\frac{p}{2}\right)^2}. \quad . \quad . \quad . \quad . \quad . \quad (72)$$

Often the work may be done more expeditiously by using the following graphical solution. Divide the net metal *ab* between rivet holes, in a horizontal direction, Fig. 52, into three equal parts. Taking a radius consisting of a rivet diameter plus $\frac{2}{3}(p - d)$, strike from each rivet center arcs intersecting at *g*. The distance *h* between *ab* and *g* will be the required distance between rows.



STAGGERED RIVET SPACING

FIG. 52.

35. Rivet Diameters. — Before taking up the design of any particular joint it is necessary to make a careful study of rivet failures. It was noted under the subject of joint failures that the destruction of a rivet might be accomplished in either of two ways, namely, by shearing or crushing. The two general expressions for rivet failure may be written:

I. When rivet shears, $Resistance = \frac{\pi d^2}{4} f_s$

II. When rivet crushes, $Resistance = dtf_c$

Assembling the constants, for a given plate thickness, and denoting them by C_1 and C_2, these expressions become

I. Shearing Resistance $= C_1 d^2$

II. Crushing Resistance $= C_2 d$

It is evident, therefore, that the shearing resistance of a rivet varies with the square of its diameter, while its crushing strength varies with the first power. A plot of shearing and crushing strengths with increasing diameters will show a curved line of parabolic character for the shearing and an inclined straight line for the crushing strength.

In Fig. 53 the curve $0a$, a parabola upon the axis $0A$, represents the increase in shearing strength of one rivet section as the diameter increases. In the same manner the curve $0b$ represents the increase in double-shearing strength, as the diameter increases. The straight lines $0c$, $0g$, etc., show the increase in crushing strength for various plate thicknesses as the diameter increases. At points such as *e*, where the crushing and shearing lines intersect, the strength of the rivet is the same for either method

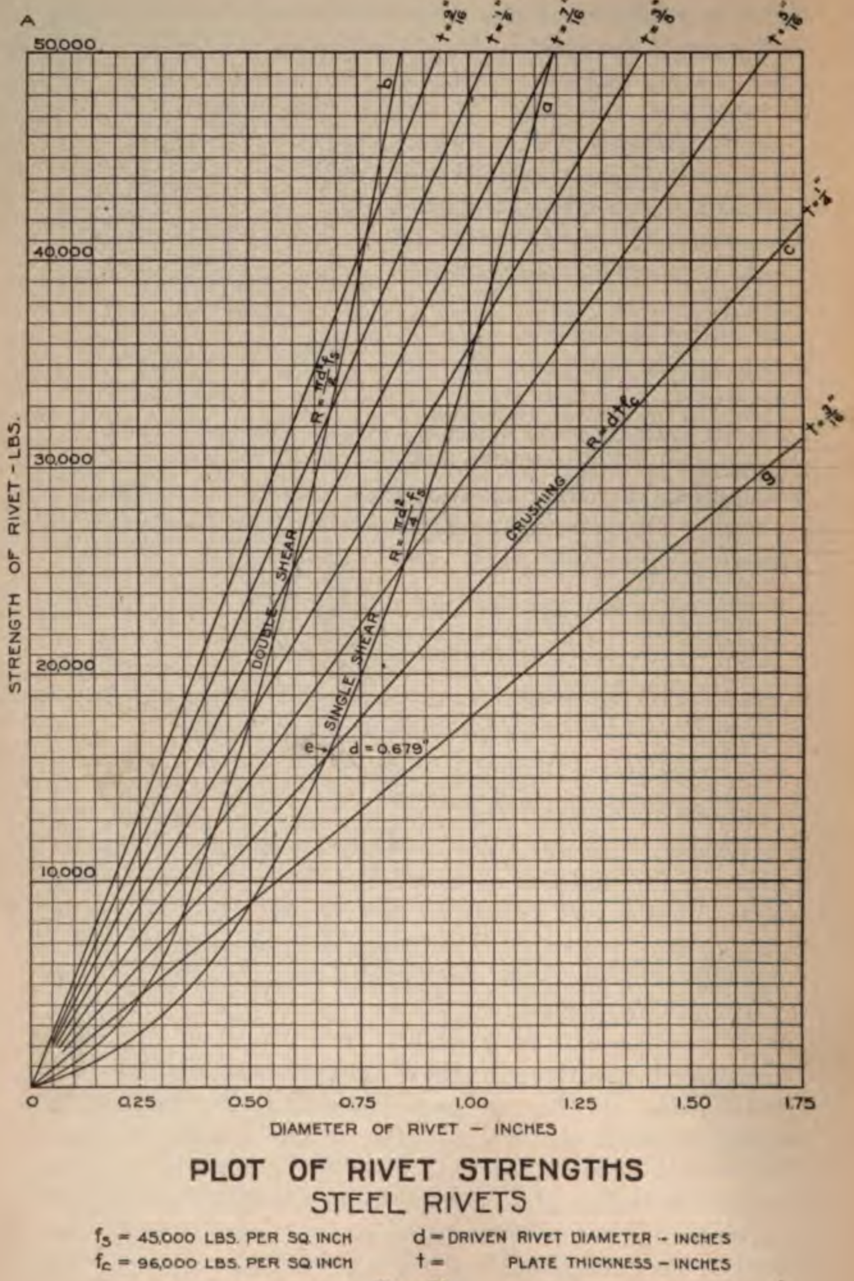

A
50,000
40,000
30,000
20,000
10,000
0
STRENGTH OF RIVET - LBS.
$t = \frac{9}{16}''$
$t = \frac{1}{2}''$
$t = \frac{7}{16}''$
$t = \frac{3}{8}''$
$t = \frac{5}{16}''$
$t = \frac{1}{4}''$
$t = \frac{3}{16}''$
b
a
c
g
e
$R = \frac{\pi d^2 f_s}{2}$
DOUBLE SHEAR
$R = \frac{\pi d^2}{4} f_s$
SINGLE SHEAR
CRUSHING $R = dtf_c$
d = 0.679"
0.25
0.50
0.75
1.00
1.25
1.50
1.75
DIAMETER OF RIVET - INCHES
PLOT OF RIVET STRENGTHS
STEEL RIVETS
f_s = 45,000 LBS. PER SQ. INCH
f_c = 96,000 LBS. PER SQ. INCH
d = DRIVEN RIVET DIAMETER - INCHES
t = PLATE THICKNESS - INCHES

Fig. 53.

of failure. Thus in $\frac{1}{4}$ in. plate a steel rivet 0.679 in. in diameter is equally liable to fail by shearing or crushing. To find this critical diameter numerically, the shearing and crushing strengths of a rivet in $\frac{1}{4}$ in. plate should be equated:

Then $$\frac{\pi d^2}{4} f_s = dtf_c,$$

or $$d = \frac{4\,tf_c}{\pi f_s}. \quad \text{. . . . . (73)}$$

From laboratory experiments upon riveted joints with steel and iron rivets the following values have been deduced:

f_s = 45,000 lbs. per sq. in. for steel rivets;
f_s = 38,000 lbs. per sq. in. for iron rivets;
f_c = 96,000 lbs. per sq. in. of projected area for steel or iron rivets in steel plates;
f_t = 55,000 lbs. per sq. in. for steel plate.

Substituting, $$d = \frac{4\,(\frac{1}{4})\,96{,}000}{(3.14)\,45{,}000} = 0.679 \text{ ins.}$$

It is evident from the plot that a rivet smaller in diameter than this critical size is liable to fail by shearing while one larger is weaker in its resistance to crushing. Hence, when the critical diameter is determined for a given thickness of plate, the method of failure of the rivets used may be predicted by reference to the curves of Fig. 53. A concrete example of values read from Fig. 53 will make the matter clearer.

Problem.

Determine the resistances to single shearing and crushing for steel rivets having diameters of $\frac{3}{4}$ in., 0.849 in., and 1 in., when used in plate $\frac{5}{16}$ in., thick.

Diameter of rivet	$\frac{3}{4}$ in.	0.849 in.	1 in.
Resistance to single shearing	19,900	25,400	35,300
Resistance to crushing	22,500	25,400	30,000

Any steel rivet, therefore, selected for use in $\frac{5}{16}$ in. plate, if less than 0.849 in. in diameter is liable to fail by shearing, and if larger, by excessive bearing pressure.

To render comparison with proposed rivet diameters more exact the following table of numerical sizes is inserted. Table II is the mathematical equivalent of the plots shown in Fig. 53. Inasmuch as soft wrought iron rivets are still used in some classes of riveted work, the table has been extended to include their critical diameters as well.

TABLE II.

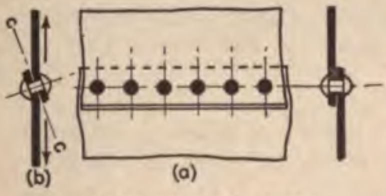
CRITICAL RIVET DIAMETERS.

Plate thickness. Ins.	Steel rivet diameters.		Iron rivet diameters.	
	Single shear. Ins.	Double shear. Ins.	Single shear. Ins.	Double shear. Ins.
$\frac{1}{4}$	0.679	0.340	0.804	0.402
$\frac{5}{16}$	0.849	0.424	1.005	0.503
$\frac{3}{8}$	1.019	0.509	1.206	0.603
$\frac{7}{16}$	1.188	0.594	1.407	0.704
$\frac{1}{2}$	1.358	0.679	1.608	0.804
$\frac{9}{16}$	1.528	0.764	1.809	0.905
$\frac{5}{8}$	1.698	0.849	2.010	1.005
$\frac{11}{16}$	1.867	0.934	2.211	1.106
$\frac{3}{4}$	2.037	1.019	2.412	1.206
$\frac{13}{16}$	2.207	1.104	2.613	1.307
$\frac{7}{8}$	2.377	1.188	2.815	1.407
$\frac{15}{16}$	2.547	1.274	3.016	1.508
1	2.716	1.358	3.217	1.608

As a summary of the above discussion the following statement may be made:

For each thickness of plate there is a critical diameter of rivet equally strong for shearing and crushing. Rivets smaller than the critical diameter fail by shearing and those larger by crushing.

36. Types of Riveted Joints. — Before proceeding to the design of riveted joints in regard to pitch values it is well to take a brief view of various forms of joints and their characteristics.



SINGLE RIVETED LAP JOINT

FIG. 54.

Single-riveted Lap Joints. — Fig. 54 represents a single-riveted lap joint. It consists merely of overlapping plates held together by one row of rivets. When exposed to tension in the direction of the arrows there is present a severe tendency to distort the joint in the manner shown. This action may be more or less relieved by bending the plates to the distorted form before riveting, but in the latter case the rivets are exposed to considerable tension as well as shearing. The bending action tends to open the laps along *cc*, producing leakage, while cracks in the plate are liable to start where the bending action is most severe.

The variation of stress in joints of this type is well illustrated by the experiments of Profs. Gibson and Barraclough to which reference was made in Fig. 14 of Chap. II. The longitudinal tension near the ring-seams of the steam drum is entirely relieved on one surface of the plate by the compression due to the distortion of the joint. Upon the reverse surface the longitudinal tension is greatly increased for the same reason. The latter fact renders this joint unfit for use in high-pressure vessels. It is, however, used for girth seams in cylindrical shells where the tension is low and where the proximity to the fire makes it necessary to employ joints with but little metal in them. The fact that such joints form part of a circular shell greatly enhances their rigidity and tends to prevent the distortion described above. With such sizes of rivets and plates as would ordinarily be used in tanks and boilers, theoretical efficiencies varying from 45 to 60 per cent may be secured with this form of joint.

Double-riveted Lap Joints. — Fig. 55 shows a double-riveted lap joint with rivets staggered. The efficiency of this type is higher than that of a single-riveted lap joint. The same tendency to distortion exists as before, however, except that the greater width of joint makes possible a preliminary alignment of plates without sharp bends. It is not, therefore, so liable to overstrain the rivets or crack the adjacent plate. This joint has considerable metal in it and hence is not adapted for girth seams over the fire. For locomotive or other internally-fired boilers where the outer shell is subjected to severe transverse stress it makes an excellent girth joint. Because of the tendency to distortion, it is rarely used for longitudinal joints under high pressures. With usual rivets and thicknesses of plate for boiler and tank work this form of joint shows efficiencies varying from 60 to 75 per cent.

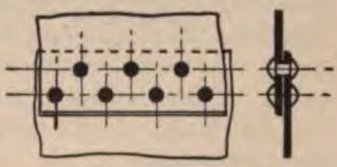


FIG. 55.

Triple-riveted Lap Joints. — Fig. 56 shows overlapping plates secured by three rows of rivets. There is here again a slight gain in efficiency over the two preceding joints. The greater width renders it easier to bring the plates into line, with the result that there is still less bending effect than in the single- and double-riveted lap joints. Triple-riveted lap joints have been used for girth seams in very heavy locomotive boilers. This joint under ordinary conditions shows efficiencies varying from 65 to 84 per cent.

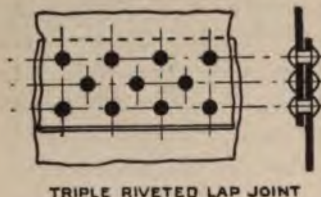
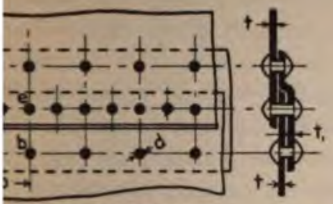


FIG. 56.

Welt-plate Joints. — In order to diminish the destructive bending tendency in lap joints without reaching the increased cost and complication of butt-joints, a form of seam was devised some years ago called a welt-plate joint. It consists of a single-, double-, or triple-riveted lap joint reinforced on the inside by a wide cover plate. The latter serves to render the joint less liable to corrosion along the lines between rivet holes, and brings about a more central loading of the joint under tension. Figs. 57 and 58 show single- and double-riveted welt-plate joints. The cover plate is bent or welted to fit the different levels of the plate, hence the name. The tendency to distort is not eliminated since the joint is not a symmetrical one. Were the upper plate, as shown in the figures, to be withdrawn from the joint by the single shearing of the rivets, the load required would be much less than if the lower one were removed accompanied by the necessary double shears. Therefore the upper side of the joint is the weaker one and the design should proceed from that standpoint. These joints were much in use for longitudinal seams on locomotive boilers some years ago, but their adaptability to the purpose has largely passed away with increasing pressures and severer demands.

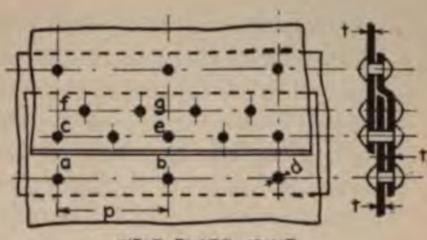
The outer rivets may be placed twice as far apart as those on the inner rows as shown in Figs. 57 and 58; or the outer spacing may be once and a half the inner. Calking along the line *ab* may then be performed easily, with the small pitch there present, while the large outer pitch brings the efficiency of this type to a much higher figure than in the case of the joints shown heretofore. Under usual conditions the theoretical efficiency of this type of joint varies from 68 to 89 per cent. This form has been largely superseded by butt-joints.

Butt-joints. — To remove effectively the tendency to distortion in riveted joints, the plates to be joined should lie naturally in



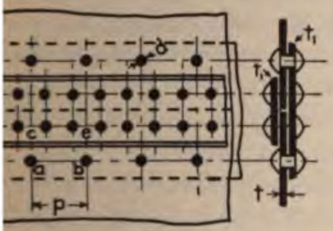
WELT PLATE JOINT
SINGLE RIVETED

FIG. 57.



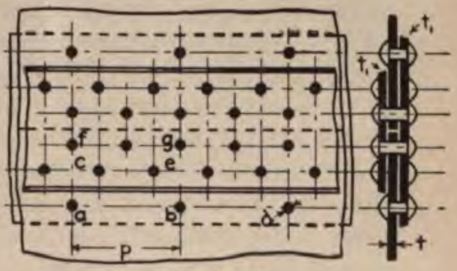
WELT PLATE JOINT
DOUBLE RIVETED

FIG. 58.



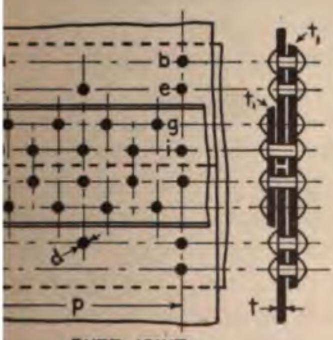
BUTT JOINT
DOUBLE RIVETED

FIG. 59.



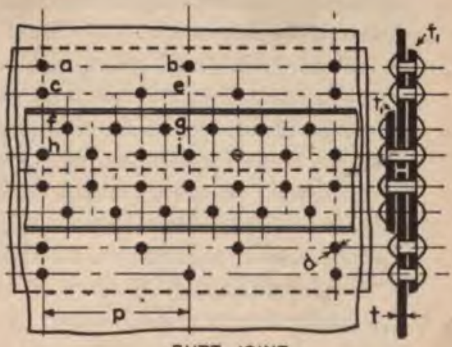
BUTT JOINT
TRIPLE RIVETED

FIG. 60.



BUTT JOINT
QUADRUPLE RIVETED

FIG. 61.



BUTT JOINT
QUADRUPLE RIVETED
ALTERNATIVE ARRANGEMENT

FIG. 62.

line with one another. Some form of cover plate or butt stra inside and outside, should then be used to complete the joi Figs. 59, 60, 61 and 62 show such types of joints.

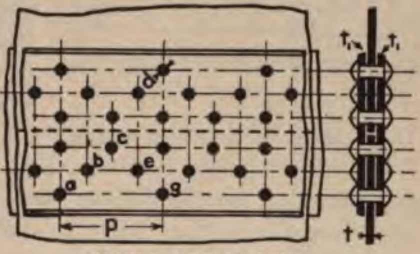
For medium pressures the butt straps may be narrow and the same width, as shown previously in Fig. 46, on page 96. O casionally when it is essential that these joints shall not off obstruction to the circulation of water in narrow spaces, the i side cover plates are omitted and the joints become single-covere butt-joints. Such joints consist essentially of two lap joints i immediate proximity to one another and are liable to the disto tion which was described above.

Where higher efficiencies are desired the inside cover may b made much wider than the outer. One, two and sometimes thre rows of rivets may be used to secure the narrow outer cover plat while one or two additional rows may be inserted in the extensio of the inside cover plate. Fig. 59 shows a typical double-rivete butt-joint; Fig. 60 a triple-riveted butt-joint; and Fig. 61 one c the quadruple-riveted pattern. As in the previous joints, th rivets on the outer row may be spaced either once and a half o twice as far apart as those on the inner row. With quadrupl riveted joints a large number of variations in rivet arrangemen is possible. One of these is illustrated in Fig. 62. These joint represent the best practice of the present time in boiler work A wide range of efficiencies varying from 75 to 95 per cent may b secured under ordinary conditions. There are very few authenti instances where longitudinal boiler seams consisting of butt joints have ruptured.

The large mass of metal in all forms of longitudinal joints re quires their total removal from the region of the fire. Very wid cover plates on the inside of cylindrical shells may tend to stretc from rivet to rivet along a cord instead of following the curvatur of the shell. When uncalked inside cover plates are thus expose to steam pressure on both sides they are termed "balanced plates, and as such do not exert the same holding power that they woul if they maintained their correct curvature. The rivet rows therefore, should not be placed unduly far apart.

In order to insure the correct curvature of the inside cove plate in butt-joints, the flat plate, after shearing to the correc size, is rolled against a heavy curved template. A series of thes templates for various boiler diameters are usually kept on han in boiler shops. By means of this operation the correct curvature of the cover plate is established throughout its entire width.

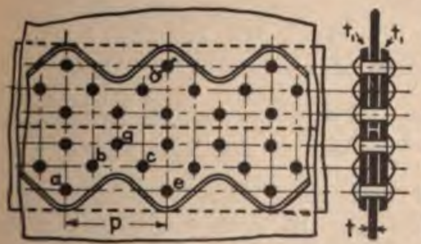
Double-shear Joints. — On account of the lack of symmetry of boiler joints having inside and outside cover plates of different widths, certain designers have recommended the use of double-shear joints similar to the one shown in Fig. 63. The outside and inside covers are generally of equal thickness and width so that the hoop tension in the shell brings a central load upon the rivets. All the rivets are in double shear, whence the name. Mr. F. W. Dean, in the Transactions of the A.S.M.E., Vol. XXXI, gives a discussion of his reasons for using this form of joint. The chief advantage consists of eliminating to a partial degree at least the uncertain strength due to curvature of the wide uncalked inside cover plate.



DOUBLE SHEAR JOINT

Fig. 63.

There are two objections to these joints. First, the efficiency is low, and second, the long reaches of plate between the rivets in the outer row are difficult to calk unless the outer cover is very heavy. In internally-fired boilers, where the thickness of the outer shell is unlimited, these joints are simple, staunch and effective, as their presence in most large Scotch marine boilers would testify. But when the use of a thin shell plate is imperative, the extended inside cover plate with the short calking pitch and high efficiency incident thereto represents the best practice. To avoid the long calking distance on the outside cover plate, the metal has sometimes been cut away as shown in Fig. 64. These scalloped edges are best shaped by the pneumatic chisel since the oxy-acetylene flame, if used, is liable to injure the quality of the

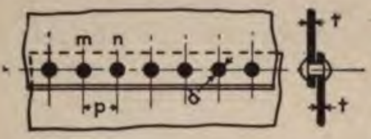


DOUBLE SHEAR JOINT
SCALLOPED COVER PLATE

Fig. 64.

steel forming the lap. Such joints show efficiencies varying from 55 to 90 per cent.

37. Calculation of Joints. — Let the simplest joint be selected as an example for calculating the various proportions. A repeating section of the single-riveted lap joint shown in Fig. 65 is comprised between the lines *m* and *n*. With adequate laps the tearing of the plate and the failure of the rivets need only be considered in this type of joint. The plates joined are supposed to be of the same thickness. Whenever one plate is thinner than the other, the joint should be designed as if for the thinner plate. Writing out the resistances to failure for section *mn* with the usual significance for f_t, f_s and f_c:



SINGLE RIVETED LAP JOINT

FIG. 65.

I. Resistance to tearing plate between holes $= (p - d)\,tf_t$

II. Resistance to shearing one rivet $= \frac{\pi d^2}{4} f_s$

III. Resistance to crushing one rivet $= dtf_c$

These three comprise the only ways in which this joint can fail. To make a perfectly designed joint the resistances to failure should all be equal. The joint would then have an exact balance of strength in all its features. Therefore the above three equations should be solved simultaneously for values of *p* and *d*.

Equating II and III

$$\frac{\pi d^2}{4} f_s = dtf_c$$

$$d = \frac{4\,tf_c}{\pi f_s}$$

This is the critical diameter of rivet equally liable to fail by crushing or shearing, and Table II, on page 104, gives a series of numerical values for this expression. Commercial rivet diameters vary by sixteenths of an inch, hence none of those shown in Table II are readily procurable. If the above theoretical rivet diameter could be used, I equated to II or III would yield a value for the pitch, and the latter would be the same from whichever equation it was derived. Imagine such a rivet to be procurable and to be used in the joint at hand. Equating I and II

$$(p - d)\, tf_t = \frac{\pi d^2}{4} f_s$$

$$p = d + \frac{\pi d^2 f_s}{4\, tf_t}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

and equating I and III

$$(p - d)\, tf_t = dtf_c$$

$$p = d\left(1 + \frac{f_c}{f_t}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad (75)$$

A numerical problem will best illustrate the above fact.

Problem.

Find the pitch value for a single-riveted lap joint using steel rivets of the critical diameter in $\frac{5}{16}$ in. plate.

From Table II the critical diameter of rivet for $\frac{5}{16}$ in. plate is 0.849 in.

Taking $f_t = 55{,}000$ lbs. per sq. in.,
$f_c = 96{,}000$ lbs. per sq. in.,
$f_s = 45{,}000$ lbs. per sq. in.,

and substituting in both equations (74) and (75), $p = 2.33$ ins.

It is interesting to introduce the quantities just determined into the original expressions for the strength of the joint and note the balance of strength. Substituting these values of pitch and rivet diameter, together with the numerical values of f_s, f_c and f_t, in the original resistances to failure, and reducing,

I. Tearing $(p - d)\, tf_t = 25{,}430$ lbs.

II. Shearing $\frac{\pi d^2}{4} f_s = 25{,}430$ lbs.

III. Crushing $dtf_c = 25{,}430$ lbs.

It is evident that a joint so designed is perfectly balanced in all its proportions, has no weak spots, and is as liable to failure in one way as another. Evidently the above procedure is purely theoretical. With commercial rivet sizes it is impossible to balance all the resistances to failure even in a joint as simple as the one under consideration. With a rivet other than the critical size, the joint will no longer be equally strong in all ways. Thus, suppose a rivet $\frac{7}{8}$ in. in diameter had been chosen. By reference to Table II it is seen that this rivet would fail by crushing.

Equating I and III, as before,

$$(p - d)\, tf_t = dtf_c$$

$$p = d\left(1 + \frac{f_c}{f_t}\right)$$

and substituting $d = \frac{7}{8}$ in. and f_c and f_t as above, the value of

$$p = 2.40 \text{ ins.}$$

The three resistances to failure will then yield numerical values as follows:

I. Tearing $(p - d)\, tf_t = 26{,}250$ lbs.

II. Shearing $\frac{\pi d^2}{4} f_s = 27{,}050$ lbs.

III. Crushing $dtf_c = 26{,}250$ lbs.

The resistances against tearing and crushing are balanced since they were equated in determining the pitch, while that of shearing shows an excess of strength due to the large rivet chosen. Had a $\frac{3}{4}$ in. rivet been chosen the manner of rivet failure would have been by shearing.

Equating, therefore, I and II, with $d = \frac{3}{4}$ in. and f_t and f_s as before,

$$(p - d)\, tf_t = \frac{\pi d^2}{4} f_s$$

$$p = d + \frac{\pi d^2 f_s}{4 f_t}$$

or

$$p = 1.91 \text{ ins.}$$

And substituting this value back in the three original resistances to failure

I. Tearing $(p - d)\, tf_t = 19{,}900$ lbs.

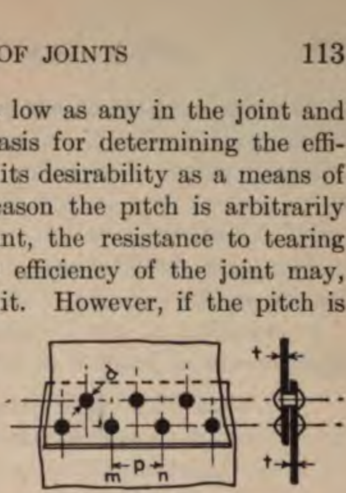
II. Shearing $\frac{\pi d^2}{4} f_s = 19{,}900$ lbs.

III. Crushing $dtf_c = 22{,}500$ lbs.

The simple principle illustrated above of throwing two features of joint strength into balance, while all the others represent excess of strength, forms the basis of the design of riveted joints. The resistance to tearing partakes in a small degree of the enhanced strength between holes, due to grooved specimen action already described in Art. 33. To a much greater degree the resistances to rivet failure are increased by the friction between the plates.

The resistance to tearing then is as low as any in the joint and should consequently be used as a basis for determining the efficiency of the joint and establishing its desirability as a means of fastening. If for some practical reason the pitch is arbitrarily reduced below the calculated amount, the resistance to tearing is still further diminished, and the efficiency of the joint may, therefore, be correctly based upon it. However, if the pitch is arbitrarily increased beyond the calculated amount, one of the rivet failures becomes the least of the resistances and the efficiency should be based upon it.

The pitch calculation for a double-riveted lap joint, Fig. 66, may be made in exactly the same manner. The resistances to failure for the repeating section *mn* are as follows:



DOUBLE RIVETED LAP JOINT

FIG. 66.

$I.$ *Tearing* $(p - d)\, tf_t$

$II.$ *Shearing* $2\left(\frac{\pi d^2}{4}\right) f_s$

$III.$ *Crushing* $2\, dtf_c$

If the joint is to unite $\frac{3}{8}$ in. plate, the critical rivet diameter from Table II is $d = 1.019$ ins. Let a $\frac{7}{8}$ in. rivet be chosen, and the rivet failure will be by shearing. The pitch formula will result from equating I and II or

$$(p - d)\, tf_t = 2\left(\frac{\pi d^2}{4}\right) f_s$$

$$p = d + \left(\frac{\pi d^2}{2\, tf_t}\right) f_s. \quad . \quad . \quad . \quad . \quad . \quad (76)$$

Had a $1\frac{1}{16}$ in. rivet been chosen, the rivet failure would have been by crushing and I = III would have yielded the pitch formula. Thus

$$(p - d)\, tf_t = 2\, dtf_c,$$

$$p = d\left(1 + \frac{2 f_c}{f_t}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (77)$$

The numerical pitch values may be found by substituting the chosen rivet diameters in formulæ (76) and (77) as before. It will be found that the gross strengths of the repeating section will exhibit the same relationship as under the analysis of the single-riveted lap joint. Another significant fact will be noted in that equations (75) and (77), expressing pitch values for crushing rivets, are independent of the plate thickness. In other words so long as the chosen rivet is large enough to fail by crushing, the pitch which follows from its use in a given joint will be the same no matter what the thickness of plate. This fact has an important bearing on the question of efficiencies as taken up later.

In summing up the foregoing it may be said that the general procedure in all joints is as follows:

1° *Write all possible methods by which the joint may fail.*
2° *Select a commercial size of rivet (from considerations yet to be explained).*
3° *By reference to Table II determine the probable method of failure of the chosen rivet.*
4° *Equate the rivet failure to the plate failure and solve for the pitch.*

38. Efficiency of Riveted Joints. — The efficiency of any joint, represented by V, is the ratio of its strength per repeating section to that of an equal length of solid plate. This ratio is usually expressed in per cent. In the analysis of the previous joint, where the pitch was calculated by the ordinary methods, the tearing strength was found to be the least of any in the joint. The remark was consequently made that it was a proper quantity on which to base the efficiency. The character of the tearing resistance is the simplest and most reliable of any in the list, since it is not involved with other resistances nor does friction between plates affect its value to any great extent. Therefore the efficiency of any joint using a pitch calculated as above will be

$$V = \frac{\textit{Weakest Resistance}}{\textit{Solid Plate Strength}} = \frac{(p-d)\,tf_t}{ptf_t}$$

$$= \frac{p-d}{p}. \quad \ldots \ldots \ldots \ldots \quad (78)$$

This is the general expression for the efficiency of any joint using the calculated pitch. If the latter is arbitrarily reduced for any reason, the tearing strength diminishes at the same time and equation (78) still represents the true efficiency. With a pitch greater than the calculated value, the rivet failures represent the lesser resistances.

Then in a single-riveted lap joint for shearing rivets, the pitch having been increased,

$$V = \frac{\frac{\pi d^2}{4} f_s}{ptf_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (79)$$

and for crushing rivets,

$$V = \frac{dtf_c}{ptf_t}$$

$$= \frac{df_c}{pf_t}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

For complicated joints using assumed pitches a general analysis of the strength must be made to discover the weakest place. The latter divided by the solid plate strength will then give the efficiency.

The end tension in a cylindrical shell is theoretically but one-half the hoop tension. Therefore, if the ring seam has half the efficiency of the longitudinal joint, there will be a fair balance of strength. The ring seams for tanks and boilers are ordinarily single-riveted lap joints. The theoretical pitch for such joints, especially when thick plate is used, may be so small that the rivet heads would be too close together for practical use. It therefore becomes necessary to increase the pitch arbitrarily, the efficiency at the same time being reduced. Let the lowest desirable efficiency be represented by $V_{(\text{min.})}$. Then for shearing rivets from equation (79),

$$V_{(\text{min.})} = \frac{\frac{\pi d^2}{4} f_s}{ptf_t}$$

or

$$p = \frac{\frac{\pi d^2}{4} f_s}{V_{(\text{min.})} tf_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

and similarly for crushing rivets,

$$p = \frac{df_c}{V_{(\text{min.})} f_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

These formulæ may be used in seeking the pitch of ring seams to give an efficiency equal to half or two-thirds that of the longitudinal joint.

The variation in efficiency for different sizes of rivets may best be studied by substituting for p in formula (78) its value as expressed in terms of shearing or crushing rivets. The pitch formula (74) for a single-riveted lap joint, shearing rivets, was

$$p = d + \frac{\pi d^2 f_s}{4\,tf_t}.$$

Substituting this value in $V = \frac{p - d}{p}$

$$V = \frac{\dfrac{\pi d^2 f_s}{4\,tf_t}}{d + \dfrac{\pi d^2 f_s}{4\,tf_t}}$$

$$= \frac{(\pi f_s)\,d}{4\,tf_t + (\pi f_s)\,d}. \quad \ldots\ldots \quad (83)$$

It is evident by inspection that this expression varies in value with d, the greater the value of the rivet diameter, the higher the efficiency. It is sometimes easier to discern this fact if the known values of π, f_s and f_t are inserted and reduced. In the latter case for steel rivets,

$$V = \frac{141.4\,d}{220\,t + 141.4\,d}. \quad \ldots\ldots \quad (84)$$

This formula is entirely equivalent to the usual efficiency formula (78) until the rivets reach a size sufficient to cause crushing. The larger the rivet, the larger the pitch and the higher the efficiency for shearing rivets. When crushing sizes are reached, the above efficiency formula (84) is no longer applicable. The pitch formula (75) for a single-riveted lap joint, crushing rivets, was

$$p = d + \frac{df_c}{f_t}.$$

Substituting the value in $V = \frac{p - d}{p}$

$$V = \frac{\dfrac{df_c}{f_t}}{d + \dfrac{df_c}{f_t}}$$

$$= \frac{f_c}{f_t + f_c}. \quad \ldots\ldots\ldots \quad (85)$$

It is evident by inspection that this expression is independent of the plate thickness or rivet diameter and is, therefore, the same for all sizes of crushing rivets. Substituting the known values of f_c and f_t

$$V = \frac{96{,}000}{55{,}000 + 96{,}000} \quad \ldots \ldots \quad (86)$$
$$= 0.636 \text{ or } 63.6 \text{ per cent.}$$

This is the uniform efficiency of single-riveted lap joints with crushing rivets. Larger rivets give larger pitches, but the efficiency remains constant. The same line of procedure with the pitch formulæ (76) and (77) for double-riveted lap joints results as follows, for shearing rivets:

$$V = \frac{(\pi f_s)\,d}{2\,tf_t + \pi f_s d} \quad \ldots \ldots \quad (87)$$

For crushing rivets,

$$V = \frac{2f_c}{f_t + 2f_c}. \quad \ldots \ldots \quad (88)$$

The latter expression reduces to $V = 77.7$ per cent, the uniform efficiency of double-riveted lap joints with crushing rivets. It will be shown later that all forms of riveted joints present a fixed value for the efficiency when crushing rivets are used.

The question of efficiencies may be summed up, then, by the following statement:

The efficiency of any riveted joint, using the calculated pitch, increases with an increasing diameter of rivets as long as the rivets shear. When a diameter sufficient for crushing has been reached, no farther increase can be secured and the efficiency is then a maximum.

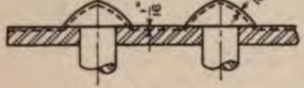
39. Factor of Safety. — Whenever the original factor of safety in the solid plate before riveting is known, the final factor may be found by multiplying the original factor by the efficiency of the joint used. But very often the original factor is lost sight of, the reason being that a commercial thickness of plate was assumed differing from the theoretical. It is then generally easier to investigate the factor independently. The factor of safety may be defined as the quotient obtained by dividing the strength of the joint at its weakest point by the load to be borne. In joints using the calculated pitch, it has been shown that $(p - d)\,tf_t$ is the lowest resistance to failure. The hoop tension per inch of length in cylindrical shells is the product of the pressure by the radius, or PR.

Then

$$\textit{Factor of Safety on longitudinal joint} = \frac{(p - d)\,tf_t}{pPR}. \quad . \; . \quad (89)$$

This is evidently equivalent to the efficiency $\frac{p-d}{p}$ multiplied by the original factor of safety $\frac{tf_t}{PR}$.

In designing shells of boilers and tanks it is well to assume a reasonable efficiency for the proposed longitudinal joint and to take this quantity into account together with the desired factor of safety in calculating the thickness. Whenever corrosion may reduce the tearing strength as shown in Fig. 67, the fact should be anticipated by using the effective section after corrosion in computing the factor of safety. When joints are protected by inside cover plates allowance for corrosion is unnecessary.



EFFECT OF CORROSION

Fig. 67.

Since in cylindrical shells without stays the end tension is but half the hoop tension, the factor of safety on ring seams will be as follows:

$$\textit{Factor of Safety on ring seam} = \frac{2\,(\textit{Efficiency of ring joint})\;tf_t}{PR}. \quad (90)$$

With stay rods or tubes the factor will be considerably more than the above expression.

40. Selection of Rivet Diameter. — From the foregoing discussion of pitch and efficiency it is evident that no absolute rule can be given for selecting the rivet size. The diameter of the rivet chosen for boiler joints depends upon four conditions:

1° *The rivet must be of commercial size, namely, its diameter must be measured in integral sixteenths.*

2° *The pitch which follows from the selected rivet size must not be so great as to permit leakage after calking.*

3° *The rivet should be such as to secure a maximum efficiency for the joint, namely, it should be a crushing rivet if possible.*

4° *The rivet must be within the capacity of the riveting machine as regards pressure when driving.*

41. Limiting Pitches. — The question of limiting pitches has been the subject of much controversy and experiment. Since very few boiler joints are calked on the inside, it is fair to assume that the steam pressure enters the joint when new to the externally calked edge. While the tightness of the rivets, due to contraction upon cooling, keeps the fluid pressure from leaking around their heads, it is certain from experiment that the fluid enters between the plates under their heads to a considerable extent. It may be assumed, as the worst possible case, that there is uniform fluid pressure throughout the joint as far as the calked edge will permit. Such a condition for a single-riveted lap joint is represented in an exaggerated form by Fig. 68. The calked edge *ab* is the final barrier to leakage and the area of plate *abce* between rivets is approximately in the condition of a beam fixed in direction at the ends and uniformly loaded. The assumption of fixed direction at the ends is warranted by the continuity of the boiler plate, and the tightness of hold which the rivets obtain by contraction.

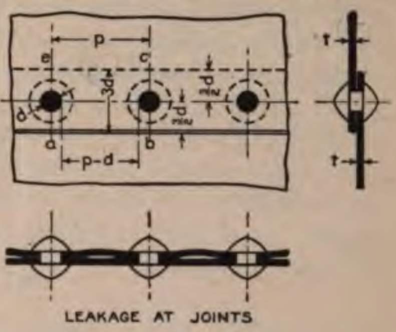


FIG. 68.

In this form of joint the plate exposed to fluid pressure is two laps wide. The usual rule that the lap is $\frac{3}{2}\,d$ would make the hypothetical beam width $3\,d$. Its length may be assumed to extend from surface to surface of rivet shank, a distance $(p - d)$, Fig. 68. Since the underlapped plate is exposed to pressure upon both sides, it will not show any deflection. The overlapped plate will bulge slightly under pressure as shown in the lower view. The usual formula for the deflection at the center of a beam fixed in direction at the ends and uniformly loaded is

$$v_0 = \frac{Wl^3}{384\,EI}.$$

The significance of the letters is as follows:

v_0 = deflection at center of beam.
W = total load distributed over the beam.
l = length of beam.
E = modulus of elasticity of beam material.
I = moment of inertia of section of beam about neutral axis.

Applying the above formula to the conditions expressed in Fig. 68, the following significance is evident:

v_0 = greatest allowable deflection at center of pitch consistent with tightness.
F = working pressure, lbs. per sq. in.
$\frac{3}{2}F$ = usual test pressure, lbs. per sq. in.
W = total load on beam area *abce* when exposed to test pressure, lbs.
$= \frac{3}{2}F(p-d)(3d)$ lbs.
$= \frac{9}{2}Fd(p-d)$ lbs.
$l = (p-d)$ ins.
E = 28,000,000 lbs. per sq. in. for steel plate.

$$I = \frac{3dt^3}{12} = \frac{dt^3}{4}.$$

Then
$$v_0 = \frac{\frac{9}{2}Fd(p-d)(p-d)^3}{384E\left(\frac{dt^3}{4}\right)},$$

or
$$v_0 = \frac{3F(p-d)^4}{64Et^3}. \qquad (91)$$

If now the maximum deflection is calculated in joints having pitches which may be considered large and near the limit, such deflections should form a fair basis for estimating the tightness of other joints. In this manner the deflection in well-designed joints has been found to approach 0.00035 in. at the center of the pitch length.

The maximum value of plate deflection between rivets consistent with tightness forms the subject of a series of tests made in 1895 by Mr. F. A. Park in the laboratories of the Massachusetts Institute of Technology. By means of very careful micrometer measurements, he found that the permissible bulging of the plate between rivets at the time leakage first appeared, lay in the region of 0.0004 in. It will, therefore, be assumed that a well-calked lap will permit a calculated deflection of 0.00035 in. at the center of the pitch length without leakage. Substituting this quantity and solving for values of $(p-d)$, equation (91) becomes

$$p - d = \sqrt[4]{\frac{64\,(28{,}000{,}000)\,(0.00035)\,t^3}{3F}}$$

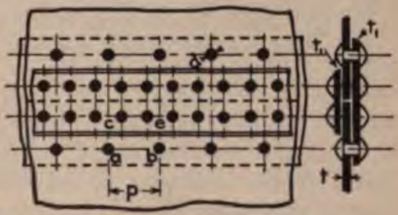
or
$$= 21.38\sqrt[4]{\frac{t^3}{F}}. \qquad (92)$$

At the end of the present chapter is given a set of tables showing the maximum allowable pitch along calked rows, based on the above discussion.

Whenever the feasibility of using a certain pitch is in question, a comparison of the proposed value with those given in Table XXIII will afford a safe criterion of calkability.

As was pointed out in a previous paragraph, crushing rivets give maximum efficiencies. Whenever consistent with tightness, therefore, it is best to select such diameters. Ordinarily hydraulic riveters will set rivets up to $1\frac{1}{4}$ ins. in diameter in a satisfactory manner. To be fully convinced that the selection of a rivet diameter is the best possible one it must be viewed from the four standpoints given above. When the maximum efficiency has been obtained, it is often best still further to increase the rivet diameter if the calking pitch will permit, in order to obtain larger pitches and, consequently, a less number of rivets in a given length of seam.

42. Method for Complicated Joints. — The double-riveted butt-joint with extended inside cover plate shown in Fig. 69 will be used to illustrate the general method of procedure for complicated joints. Similar methods may be pursued for other types. In the joint chosen each cover plate bears approximately half the load. To avoid unnecessary complication, the sum of the cover plate thickness is made considerably greater than the boiler plate thickness. Therefore, the cover plates, even after corrosion, will never tear. In the following analysis of the joint, a method will be shown for calculating the least allowable cover plate thickness. Very frequently, in boiler work, both covers are made $\frac{1}{16}$ in. less in thickness than the main plate. Whenever there is difficulty in calking the outer cover to tightness, the thickness of the latter may be made equal to or $\frac{1}{16}$ in. greater than that of the main plate. For convenience a repeating section *ab* will be so chosen as to embrace one pitch length of plate on the outer row of rivets. The resistances to failure will be taken up, first, in regard



BUTT JOINT
DOUBLE RIVETED

FIG. 69.

to tearing the plate, and second, from the standpoint of rivet failure.

I. Tearing on line ab.

Resistance $= (p - d)\,tf_t$

This is the simplest and most reliable of all the resistances. For this reason it is kept the lowest and is later made the basis of the efficiency calculation. A well-designed joint should have the other resistances, working inward, in a gradually increasing scale of value.

II. Tearing on line ce combined with shearing or crushing the outer rivet on line ab.

For shearing rivets.

(*a*) *Resistance* $= (p - 2\,d)\,tf_t + \frac{\pi d^2}{4} f_s$

For crushing rivets.

(*b*) *Resistance* $= (p - 2\,d)\,tf_t + dt_1 f_c$

Inasmuch as there are more holes on the inner line than the outer, there will be less plate strength on line *ce*, hence tearing on the inner row must be investigated. The extended inside cover plate provides a secure reinforcement against tearing on *ce* by the holding power of the rivet at *ab*. Before the boiler plate can be torn at *ce*, the holding power of the outer rivet must evidently be destroyed by shearing or crushing. Since the outer rivet joins plates of two different thicknesses, the crushing will take place against the thinner one. Therefore, t_1 is used in II (*b*).

Sometimes an additional row of rivets is inserted in the narrow cover plate making a triple-riveted butt-joint, Fig. 60, on page 107. It may be noted that tearing upon the inmost line of rivets is never liable to occur. The tearing areas of plate per repeating section on lines *ce* and *fg* are equal. Before tearing can take place on *fg*, all the rivets outside of that line must be destroyed. Manifestly the inmost row presents a large excess of strength over the two outer ones and no additional knowledge will be gained by including its strength in the list of resistances to failure.

Next, rivet failures will be considered.

III. The rivets may all shear.

Resistance $= 5\left(\frac{\pi d^2}{4}\right) f_s$

Two of the rivets are in double shear and one in single. In this discussion double shear will be taken equal to twice single shear, for reasons previously stated.

IV. The rivets may all crush.

Resistance $= 2\,dtf_c + dt_1 f_c$

The inner rivets receive the full bearing pressure of the boiler plate while the outer one, joining as it does a thin cover to a thick plate, would be the more severely stressed by the thin plate. Hence the crushing of the outer rivet will take place against the plate of thickness t_1 as in II (*b*).

V. The rivets on the inner row may crush and those on the outer row shear.

(*a*) *Resistance* $= 2\,dtf_c + \frac{\pi d^2}{4} f_s$

This manner of failure is worthy of consideration, because of the difference in the conditions surrounding the two lines of rivets. The inner rivets, securely held by the cover plates, are in good condition to crush, while the outer ones may very probably shear. In certain forms of welt joints, Figs. 57 and 58, on page 107, the manner of failure is just the reverse, that is, the inner rivets shear while the outer ones crush.

(*b*) *Resistance* $= 4\left(\frac{\pi d^2}{4}\right) f_s + dt_1 f_c$

This resistance is made possible by the fact that the inner rivets join thick plates, while the outer ones receive the bearing pressure of the thin cover, and hence are liable to crush.

The above five resistances complete the possible ones for these types of joints. All the resistances involving failure of rivets have a large increment of frictional resistance to be added to the expressions above. While the amount of this resistance is a matter of conjecture, it is, nevertheless, a real and potent factor in diminishing the liability to rivet failure. It is necessary before calculating the pitch to insure an increase of strength on the inner lines of the joint. The necessary conditions may be determined mathematically by making

$$II\ (a) > I \text{ and } II\ (b) > I.$$

That is, $$(p - 2\,d)\,tf_t + \frac{\pi d^2}{4} f_s > (p - d)\,tf_t$$

Simplifying $$d > \frac{4\,tf_t}{\pi f_s}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9$$

Again $$(p - 2\,d)\,tf_t + dt_1f_c > (p - d)\,tf_t$$

Simplifying $$t_1 > \frac{tf_t}{f_c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9$$

In this manner two arbitrary limits are set up, one for rivet si and one for cover plate thickness, which must be heeded in ord to insure an increase in the strength of the joint upon the inn rows. With practical values for rivet diameter and cover pla thickness, the above limits are rarely of much importance, sin the sizes actually used generally exceed those theoretically r quired. In rare instances, where the plates to be joined are ve thin, the theoretical cover plate thickness may fall below half t main plate thickness. In such cases, the covers should each made to exceed the half plate thickness by a liberal margin.

The determination of the above limits is all that resistance II (and II (*b*) can establish, the pitch disappearing by cancellation the inequalities. To calculate the pitch; first, select a rivet larg than the limit just established; second, eliminate by inspecti the resistances as to rivet failure which represent excess of strengt and lastly, equate the one remaining rivet resistance to the teari of the plate and solve for the pitch. A numerical example w best illustrate the procedure.

Problem.

Let the boiler plate in the joint of Fig. 69 be $\frac{3}{8}$ in. thick and both covers $\frac{1}{16}$ less. Then $t = \frac{3}{8}$ in. and $t_1 = \frac{5}{16}$ in. A reference to Table II, page 1 gives $d_1 = 0.509$ in. as the critical steel rivet diameter for equal crushing a double shearing in $\frac{3}{8}$ in. plate, and $d_2 = 0.849$ in. as the critical steel ri diameter for equal crushing and single shearing in $\frac{5}{16}$ in. plate.

The solution of equation (93) gives a limiting minimum rivet diameter equal to 0.58 in. Equation (94) yields a minimum cover plate thickness 0.21 in. Then, the assumed cover plates $\frac{5}{16}$ in. thick are suitable ones an rivet larger than 0.58 in. such as $\frac{7}{8}$ in., for instance, is a proper one. Compari $\frac{7}{8}$ in. with $d_1 = 0.509$ in. and $d_2 = 0.849$ in., the method of rivet failure may seen to be crushing throughout the joint, since d is larger than both d_1 and All the resistances to rivet failure except IV, therefore, may be eliminated fr further consideration. A pitch value determined by making I = IV w throw these two strengths into balance and all the other resistances to failu will represent excess strengths.

Now the friction between plates increases IV far beyond its mathemati value while I has only a slight enhancement of value due to grooved spe men action. Resistance I is, therefore, the least in the list and a proper one on which to base the efficiency.

Equating I and IV

$$(p - d)\,tf_t = 2\,dtf_c + dt_1f_c.$$

Substituting

$$d = \tfrac{7}{8}\text{ in.},\quad t = \tfrac{3}{8}\text{ in.},\quad t_1 = \tfrac{5}{16}\text{ in.}$$

and the usual values for f_t and f_c,

$$p = 5.20\text{ ins.}$$

It is interesting to substitute back the values just determined in all five of the resistances and note the relative strengths of a repeating section of the joint. The following is the numerical result:

I. 89,240 lbs.
II. (*a*) 98,210 lbs.
II. (*b*) 97,400 lbs.
III. 135,300 lbs.
IV. 89,240 lbs.
V. 90,050 lbs.

Having calculated the pitch, it yet remains to show that it is a proper one to use from the three following standpoints:

(*a*) *Calkability.*
(*b*) *Efficiency.*
(*c*) *Factor of Safety.*

The calking on this joint is performed along the inside line *ce* and the length of plate to be compared with the values in Table XXIII, at the end of the chapter, is

$$\frac{p}{2} - d = 1.725\text{ ins.}$$

A reference to the outer cover plate thickness and test pressure at hand will establish the possibility of using the above pitch.

The efficiency and factor of safety should next be investigated under the principles laid down in Arts. 38 and 39. Early in the discussion of riveted joints it was stated that, whenever the rivets failed throughout the joint by crushing, the efficiency was constant and a maximum. The general expression for the efficiency of a riveted joint, using the calculated pitch, was previously shown to be

$$V = \frac{p - d}{p}.$$

Substituting the value for numerator and denominator from th pitch formula used in the joint under discussion,

$$V = \frac{\dfrac{2\,dtf_c + dt_1f_c}{tf_t}}{\dfrac{2\,dtf_c + dt_1f_c + dtf_t}{tf_t}}$$

$$= \frac{2\,tf_c + t_1f_c}{2\,tf_c + t_1f_c + tf_t}. \quad . \; . \; . \; . \; . \; . \; . \; . \quad (95)$$

It is evident upon inspection that this expression is a constant for a given thickness of main and cover plate. If the cover plate on the inside of the joint is made the same thickness as the main plate, the expression becomes

$$V = \frac{3f_c}{3f_c + f_t}, \quad . \; . \; . \; . \; . \; . \; . \; . \quad (96)$$

which is the maximum efficiency obtainable for this type of joint. Substituting numerical values

$$V = \frac{3\,(96{,}000)}{3\,(96{,}000) + 55{,}000}$$

$$= 84.0 \text{ per cent.}$$

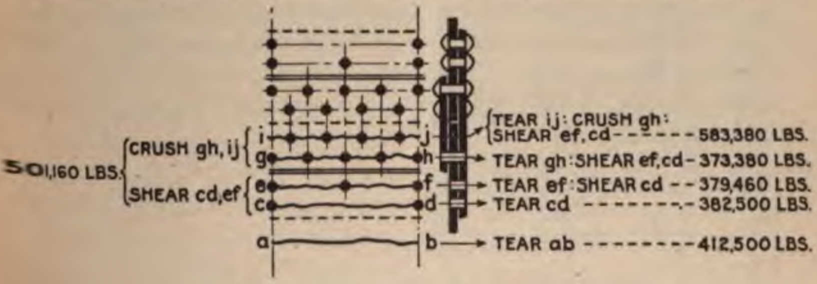
Hence, in this case the above statement in regard to constant efficiency is shown to be true. A reference to the tables of pitches at the end of this chapter also proves that the efficiency with crushing rivets is a maximum. With the inside cover plate somewhat thinner than the main plate, the efficiency is reduced a little in its value. A higher efficiency must be sought by using a more complicated type of joint.

43. General Efficiency Calculation. — When the pitch of a riveted joint has been assumed from entirely arbitrary reasons, the methods previously outlined for determining the efficiency are not applicable. A general analysis of the joint strength must be made. The ratio of the least resistance to failure to the strength of the solid plate will then constitute the efficiency. The manner of failure of each row of rivets can be predicted by reference to Table II, page 104. The strength of a repeating section of the joint should next be calculated for each row of rivets, passing from the outside of the joint inward. The lowest resistance to failure can then be detected without question. Let the quadruple-riveted butt-joint of Fig. 70 represent the proportions of such an arbitrary design. Let the long pitch be taken as 12 ins., the main plate $\frac{5}{8}$ in. thick, the inside cover $\frac{3}{8}$ in. thick, the outside cover $\frac{1}{2}$ in. thick and the rivets $\frac{7}{8}$ in. in diameter. In the figure the strength along each line of rivets is placed at the side and the ratio

$$\frac{373{,}380}{412{,}500} = 0.905 \quad \text{or} \quad 90.5 \text{ per cent,}$$

represents the efficiency of the joint.

44. Minimum Pitch. — Sometimes instead of using a theoretically correct pitch as calculated above, a considerably smaller distance between rivets is arbitrarily chosen. There is danger in so doing that the tearing strength will be unduly lowered on the inner row of rivets. In the numerical example given in Art. 42, the rivets failed by crushing throughout the joint. The possible failures were, then,


501,160 LBS.
CRUSH gh, ij
SHEAR cd,ef
i
g
e
c
j
h
f
d
a
b
TEAR ij: CRUSH gh: SHEAR ef, cd - - - - - - 583,380 LBS.
TEAR gh: SHEAR ef, cd - 373,380 LBS.
TEAR ef: SHEAR cd - - 379,460 LBS.
TEAR cd - - - - - - - - - 382,500 LBS.
TEAR ab - - - - - - - - - 412,500 LBS.

STRENGTH OF JOINT

FIG. 70.

I. *Tearing outside.*
II. (*b*) *Tearing and crushing.*
IV. *Crushing all rivets.*

Now I = IV gave the correct pitch, after having made II (*b*) > I by use of a suitable cover plate. Supposing the correct pitch to be arbitrarily diminished, the question arises as to how far the reduction may wisely be carried. Making the strength on the inner row greater than the rivet failure may be expressed by the relation

$$II\ (b) > IV.$$

Then

$$(p - 2\,d)\,tf_t + dt_1f_c > 2\,dtf_c + dt_1f_c$$

or

$$(p - 2\,d)\,tf_t > 2\,dtf_c$$

This equation may be interpreted to mean that there should always be enough tensile strength of plate between rivet holes in well-designed joints to load the rivets to their full shearing or crushing values. If insufficient plate were used, the rivets would not participate fairly in sustaining the load on the joint.

Solving and substituting

$$t = \tfrac{3}{8} \text{ in.}, \quad d = \tfrac{7}{8} \text{ in.},$$

and the usual constants for f_t and f_c

$$p > 4.80 \text{ ins.}$$

With this value used as the pitch, I will be less than II (*b*), and II (*b*) will equal IV. A further reduction will make II (*b*) less than the rivet strength IV. Therefore, by equating II (*a*) or (*b*) to the rivet strength a minimum pitch may be found, below which it will not be wise to go. This can readily be seen from a numerical comparison. With $p = 4.80$ ins. the strength of the joint per repeating section is

I. = 80,970 lbs.
II. (*b*) = 89,210 lbs.
IV. = 89,210 lbs.

Any less pitch than the above minimum one will make

$$II\ (b) < IV,$$

and the full rivet strength will not be reached, a result not consistent with good design.

45. General Procedure. — Before dismissing the subject of pitch calculation, the method of procedure for joints in general will be summarized:

Method of Procedure

1° *Select type of joint.*
2° *Write resistances to failure:*

Plate
- *I.* *Tearing outside row.*
- *II.* (*a*) *Tearing inside row and shearing outer rivet.*
- *II.* (*b*) *Tearing inside row and crushing outer rivet.*

Rivet
- *III.* *Shearing all rivets.*
- *IV.* *Crushing all rivets.*
- *V.* (*a*) *Combination of rivet failures by crushing*
- *V.* (*b*) *and shearing on inner and outer rows.*

3° *Find limits:*

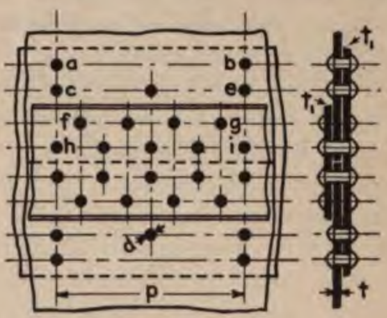
$II\,(a) > I$ *Find min.* d.
$II\,(b) > I$ *Find min.* t_1

4° *Select trial rivet diameter and predict failure by reference to Table II.*

5° *Eliminate excess rivet strengths by inspection and equate remaining one to I for pitch value.*

6° *Determine suitability of pitch from standpoint of calkability, efficiency, and factor of safety.*

46. Butt-Joints of High Efficiency. — When extra high pressures are used in boilers of large diameter the efficiency of the longitudinal seams may be considerably increased by the use of quadruple-riveted butt-joints, Fig. 71. It may again be emphasized that the large mass of metal in such joints renders imperative their removal from the region of the fire, while the necessarily wide inside cover plate forbids their use except in boilers of large diameter. The design of such joints follows the general method outlined for ordinary joints. The resistances to failure are as follows:



BUTT JOINT
QUADRUPLE RIVETED

FIG. 71.

I. Tearing the boiler plate on ab.

$$Resistance = (p - d)\,tf_t$$

II. The next inner row ce has one additional hole in it, hence its tearing strength coincident with the outer rivet failure must be included in the resistances.

$$(a)\ Resistance = (p - 2\,d)\,tf_t + \frac{\pi d^2}{4} f_s$$

$$(b)\ Resistance = (p - 2\,d)\,tf_t + dt_1 f_c$$

III. The next inner row fg has two more holes than those already considered. The holding power of the outer rivets has increased, however.

$$(a)\ Resistance = (p - 4\,d)\,tf_t + 3\,\frac{\pi d^2}{4} f_s$$

$$(b)\ Resistance = (p - 4\,d)\,tf_t + 3\,dt_1 f_c$$

The fourth row *hi* presents no change in tearing strength other than those already considered. The rivets in rows *ab* and *ce* are under identical conditions as regards plate thickness, hence their crushing or shearing may be taken in the aggregate.

IV. All rivets shear.

$$\text{Resistance} = 19\left(\frac{\pi d^2}{4}\right) f_s$$

V. All rivets crush.

$$\text{Resistance} = 8\,dtf_c + 3\,dt_1 f_c$$

VI. The two inner rows of rivets may crush and the two outer ones shear.

$$\text{Resistance} = 8\,dtf_c + 3\left(\frac{\pi d^2}{4}\right) f_s$$

To preserve the increased strength along the rivet lines as the center line of the joint is approached, the following inequalities must be considered:

$$\begin{aligned} II\,(a) &> I,\\ II\,(b) &> I,\\ III\,(a) &> II\,(a),\\ III\,(b) &> II\,(a),\\ III\,(a) &> II\,(b),\\ III\,(b) &> II\,(b). \end{aligned}$$

Writing and reducing these inequalities,

$$(p - 2\,d)\,tf_t + \left(\frac{\pi d^2}{4}\right) f_s > (p - d)\,tf_t$$

$$\therefore\ d > \frac{4\,tf_t}{\pi f_s}. \qquad (97)$$

$$(p - 2\,d)\,tf_t + dt_1 f_c > (p - d)\,tf_t$$

$$\therefore\ t_1 > \frac{tf_t}{f_c}. \qquad (98)$$

$$(p - 4\,d)\,tf_t + 3\left(\frac{\pi d^2}{4}\right) f_s > (p - 2\,d)\,tf_t + \left(\frac{\pi d^2}{4}\right) f_s$$

$$\therefore\ d > \frac{4\,tf_t}{\pi f_s}. \qquad (99)$$

$$(p - 4\,d)\,tf_t + 3\,dt_1 f_c > (p - 2\,d)\,tf_t + \frac{\pi d^2}{4} f_s$$

The latter expression involves the shearing and crushing of the same rivet under similar conditions. The only rivet which could do this would be one of exactly the limiting diameter. Under such conditions

$$\frac{\pi d^2}{4} f_s = d t_1 f_c$$

and the above equation becomes

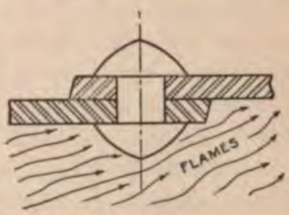
$$2\,d t_1 f_c > 2\,d t f_t$$

or

$$t_1 > \frac{t f_t}{f_c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

The inequalities, III $(a) >$ II (b) and III $(b) >$ II (b), yield the same limiting values for d and t_1. Expressions (97), (98), (99) and (100) are the same as those under the former joint and afford the usual limits for cover plate thickness and rivet diameter. In certain joints of this class with the rivets spaced as in Fig. 62, page 107, there are two limiting rivet diameters determined by equations (97) and (99) and two limiting cover plate thicknesses following from (98) and (100). The procedure is the same, however. Having selected a suitable rivet diameter and cover plate thickness in the light of the above equations, the resistances representing excess rivet strengths may be eliminated by inspection and the remaining one equated to the tearing of the plate to find the pitch. The investigation of calkability, efficiency and factor of safety is pursued as before.

47. Fire Seams. — The preceding discussion of riveted joints has reference to seams so far removed from the fire as to be subjected only to the heat of the fluid confined within the boiler. Heat of such a limited degree does not seriously affect the calking of the joint. When, however, seams are directly exposed to the fire, the previous discussion of pitch limits is no longer applicable. The joint selected for such use is always of the single-riveted lap type. This form presents a minimum amount of metal to the destructive action of the fire. The overlap of the plates should be as small as is consistent with the necessary strength and the arrangement of the joint should be such that the flames will not impinge directly against the outside lap. Fig. 72 shows the proper arrangement with regard to the direction of the flames. The same size of rivets should

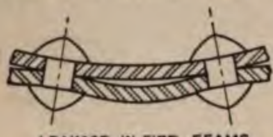


FIRE SEAM

FIG. 72.

be used in the ring seams as in the longitudinal ones, to avoid confusion at the junction of the two.

However perfect the circulation within the boiler, the outer plate in the ring seam is more or less overheated where the latter passes over the fire. There is, therefore, a tendency for the outer plate to expand away from the inner one and thereby to cause leakage. Fig. 73 shows this tendency in an exaggerated manner. For this reason the value of $(p - d)$ on fire seams must be kept very much lower than on cool seams. The following arbitrary limits have been established by long practice as safe ones:



LEAKAGE IN FIRE SEAMS

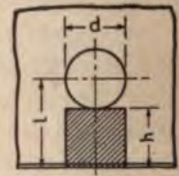
Fig. 73.

VALUES FOR $(p - d)$ IN THE RING SEAMS OF EXTERNALLY-FIRED BOILERS.

Thickness of plate. In.	Value of $(p - d)$ for hot seams. Ins.
$\frac{1}{4}$	$1\frac{1}{4}$
$\frac{5}{16}$	$1\frac{1}{4}$
$\frac{3}{8}$	$1\frac{5}{16}$
$\frac{7}{16}$	$1\frac{3}{8}$
$\frac{1}{2}$	$1\frac{3}{8}$
$\frac{9}{16}$	$1\frac{3}{8}$

Inasmuch as there is always some excess circumference of plate in the outer course, ring seams should not be riveted up pitch after pitch in one direction. Four or six rivets should be driven at the start, equally spaced around the seam, in order to distribute any slack plate as uniformly as possible. For ease also in laying out the ring seam, the number of spaces should be a multiple of four or six, if possible.

48. Lap Calculation. — While the margin of plate l, Fig. 74, from edge to center of rivet hole is often arbitrarily assumed to be one and one-half times the driven rivet diameter, its value is nevertheless a very important factor in the success of a riveted joint. As was briefly pointed out early in the discussion of riveted joints, Art. 29, page 93, the function of the lap is twofold. Its office in pressure work is, first, to provide a stiff piece of plate which, when initially sprung back by calking, will react against the other plate of the joint with sufficient intensity to prevent the escape of the fluid. The second object of the lap is to throw



LAP CALCULATION

Fig. 74.

across in front of the rivets a piece of plate of width sufficient to bring the full rivet strength into play.

The second criterion, that of strength, is the one best fitted for consideration in lap design. To find the load actually imposed upon the lap, taking into account all the peculiarities of various joints, would only yield a vast amount of unwarranted complication. Hence, in general, each rivet and the lap ahead of it, will be isolated from the rest of the joint. The lap must be strong enough to resist the greatest load the rivet can impose upon it. In this work the metal ahead of the rivet, shown section-lined in Fig. 74, will be considered to constitute a beam of length d and height h. From the continuity of the plate the lap-beam will be assumed to be fixed in direction at its ends. The distribution of the rivet load over the plate ahead of it is a matter of some conjecture and the stresses are very complex. It has been found by experiment to yield good practical results if the rivet load is taken as concentrated at the center of the lap-beam. The bending moment at the center of a beam so loaded is

$$M = \frac{WL}{8},$$

the letters having the usual significance. The load which the rivet can impose upon the lap-beam will depend upon the manner of rivet failure. Three cases may be considered:

I. Shearing the rivet once,

$$Load = \frac{\pi d^2}{4} f_s$$

II. Shearing the rivet in double shear,

$$Load = \frac{\pi d^2}{2} f_s$$

III. Crushing the rivet or plate,

$$Load = dtf_c$$

These three constitute all possible rivet loads.

The peculiar features of two joints may also be considered as having a bearing upon the lap calculation. In double-covered butt-joints each cover plate holds approximately half the boiler plate tension, Fig. 46, page 96. The rivet then constitutes a beam supported at the ends by the covers and loaded at the center by the boiler plate. The inner cover may be of the extended type, in which case its increased width will take care of the above rivet load. The narrow outer cover will be liable to lap failure unless calculated to withstand the reaction at the outer end of the rivet. In case the rivet is small enough to fail by double shearing, the outer cover plate lap will be loaded with the shearing of one rivet section and the case reverts to I, considered above. With large rivets the failure may result in crushing. Each cover plate will then receive as a maximum load one-half the rivet's crushing strength or,

IV. Half-crushing,

$$Load = \frac{dtf_c}{2}$$

When the pitch in the single-riveted lap joints, Fig. 65, page 110, is arbitrarily diminished for reasons previously described under fire seams, the rivets are excessively strong, and, before the full rivet strength is reached, such a joint would have ruptured by tearing. It is, therefore, unnecessary to make the lap strong enough to hold the full rivet strength. If the lap will hold the diminished tearing strength of the plate, it will be as good as the rest of the joint. Under such conditions the load becomes

V. Tearing between rivets,

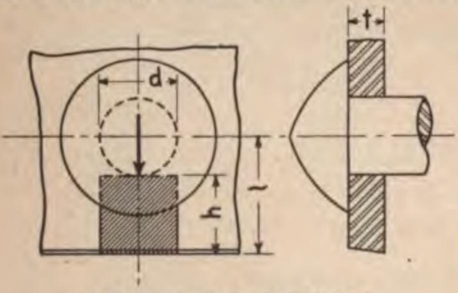
$$Load = (p - d)\, tf_t$$

To simplify the work of lap calculation the above five loads will be substituted in the usual beam formula and the resulting equations reduced to a simple form. The formula,

$$M = f\frac{I}{y},$$

when applied to a beam of rectangular section with the dimensions given in Fig. 75, becomes

$$M = f\frac{th^2}{6}.$$



LAP CALCULATION

Fig. 75.

The maximum bending moment for a beam fixed in direction at the ends and loaded with a concentrated load at the center is, for this case,

$$M = \frac{(Load)\,(d)}{8}.$$

For a ductile material like boiler plate it may be safely assumed that the modulus of rupture is approximately equal to the tensile strength, or f = 55,000 lbs. per square inch. Then

$$\frac{(Load)\,(d)}{8} = f_t \frac{th^2}{6}$$

or

$$h = \sqrt{\frac{3}{4}\frac{d}{f_t t}\,(Load)}$$

$$= 0.00369\sqrt{\frac{d}{t}\,(Load)}.$$

The load expressions in the five cases cited above may next be substituted. Inserting at the same time the value of π, the usual values of f_s and f_c and remembering that the lap is measured to the center of the rivet, the expressions become

I. Single shearing.

$$Lap = \frac{d}{2} + 0.00369\sqrt{\frac{d}{t}\left(\frac{\pi d^2}{4}\right)f_s}$$

For steel rivets,

$$Lap = \frac{d}{2} + 0.694\sqrt{\frac{d^3}{t}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (101)$$

II. Double shearing.

$$Lap = \frac{d}{2} + 0.00369\sqrt{\frac{d}{t}\left(\frac{\pi d^2}{2}\right)f_s}$$

For steel rivets,

$$Lap = \frac{d}{2} + 0.982\sqrt{\frac{d^3}{t}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (102)$$

III. Crushing.

$$Lap = \frac{d}{2} + 0.00369\sqrt{\frac{d}{t}\,(dtf_c)}$$

$$= 1.645\,d. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (103)$$

This is independent of the plate thickness, hence with a given crushing rivet the lap is the same for all thicknesses of plate.

IV. Half-crushing.

$$Lap = \frac{d}{2} + 0.00369\sqrt{\frac{d}{t}\left(\frac{dtf_c}{2}\right)}$$

If the cover plate thickness is the same as that of the boiler plate this expression becomes

$$Lap = 1.309\,d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (104)$$

an expression like equation (103) independent of the plate thickness. When the thickness of the cover, t_1, is different from that of the boiler plate, as is usually the case, the above general expression becomes

$$Lap = \frac{d}{2} + 0.00369 \sqrt{\frac{d}{t_1}\left(\frac{dtf_c}{2}\right)}$$

$$= \frac{d}{2} + 0.809\, d \sqrt{\frac{t}{t_1}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (105)$$

V. Tearing.

$$Lap = \frac{d}{2} + 0.00369 \sqrt{\frac{d}{t}(p - d)\, tf_t}$$

$$= \frac{d}{2} + 0.866 \sqrt{(p - d)\, d}. \quad . \quad . \quad . \quad . \quad (106)$$

This is again independent of the plate thickness, hence for a given pitch and rivet the lap is the same for all thicknesses of plate provided a reduced pitch is used.

In all cases, the lap must exceed the rivet diameter by a reasonable amount in order to provide a good bearing for the rivet heads. The latter approach 2 d in diameter as was shown under rivet proportions.

In order to render formulæ (101) to (106) inclusive as useful as possible in practice, the following plots and tabulations have been calculated for the usual range of rivet diameters and plate thicknesses. The results of formulæ (101), (102) and (103), for single and double shearing and crushing are contained in Tables III and IV. Whenever the rivet is large enough to fail by crushing rather than shearing, the fact is indicated by the style of type as explained under the column headed "Remarks." After having selected a lap value for a given case, it should generally be expressed to the next integral sixteenth of an inch above the decimal values given in the tables.

The application of formulæ (104) and (105) for half crushing laps is represented by the plot of Fig. 76. For abscissæ the varying ratios of the main plate thickness to that of the cover plate are plotted while the ordinates comprise the corresponding lap values. For equal main and cover plate thicknesses it is evident that the lap equals 1.309 d in each case as given in formula (104). The plotted lines are continued to the left until the indicated lap value is but $\frac{1}{8}$ in. greater than the half rivet head. Laps less than these do not find practical use.

TABLE III.

CALCULATED LAP VALUES FOR STEEL RIVETS.

Single Shearing and Crushing.

f_s = 45,000 lbs. per sq. in. f_c = 96,000 lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Thickness of plate forming lap. Ins.
$\frac{5}{8}$	*1.00*	*0.93*	*0.87*	*0.83*	*0.80*	*0.77*	*0.75*	....	....	....	....	....	....	Remarks. The following style of figures indicates failure by single shearing. *1 2 3 4 5 6 7 8 9 0* The following style of figures indicates failure by crushing. 1 2 3 4 5 6 7 8 9 0
$\frac{11}{16}$	1.13	*1.05*	*0.99*	*0.94*	*0.90*	*0.87*	*0.84*	*0.82*	....	....	....	....	....	
$\frac{3}{4}$	1.23	*1.18*	*1.11*	*1.06*	*1.01*	*0.98*	*0.95*	*0.92*	*0.90*	*0.88*	....	....	....	
$\frac{13}{16}$	1.34	*1.32*	*1.24*	*1.18*	*1.13*	*1.08*	*1.05*	*1.02*	*0.99*	*0.97*	*0.95*	....	....	
$\frac{7}{8}$	1.44	1.44	*1.37*	*1.30*	*1.24*	*1.20*	*1.16*	*1.12*	*1.09*	*1.07*	*1.04*	*1.03*	*1.01*	
$\frac{15}{16}$	1.54	1.54	*1.50*	*1.42*	*1.36*	*1.31*	*1.26*	*1.23*	*1.20*	*1.17*	*1.14*	*1.12*	*1.10*	
1	1.65	1.65	*1.63*	*1.55*	*1.48*	*1.43*	*1.38*	*1.34*	*1.30*	*1.27*	*1.24*	*1.22*	*1.19*	
$1\frac{1}{16}$	1.75	1.75	1.75	*1.68*	*1.61*	*1.55*	*1.49*	*1.45*	*1.41*	*1.38*	*1.34*	*1.32*	*1.29*	
$1\frac{1}{8}$	1.85	1.85	1.85	*1.81*	*1.73*	*1.67*	*1.61*	*1.56*	*1.52*	*1.48*	*1.45*	*1.42*	*1.39*	
$1\frac{3}{16}$	1.95	1.95	1.95	*1.94*	*1.87*	*1.79*	*1.73*	*1.68*	*1.63*	*1.59*	*1.56*	*1.52*	*1.49*	
$1\frac{1}{4}$	2.06	2.06	2.06	2.06	*2.00*	*1.92*	*1.85*	*1.79*	*1.75*	*1.70*	*1.66*	*1.63*	*1.60*	
$1\frac{5}{16}$	2.16	2.16	2.16	2.16	*2.13*	*2.05*	*1.98*	*1.92*	*1.86*	*1.81*	*1.77*	*1.73*	*1.70*	
$1\frac{3}{8}$	2.26	2.26	2.26	2.26	2.26	*2.18*	*2.10*	*2.04*	*1.98*	*1.93*	*1.88*	*1.84*	*1.81*	

TABLE IV.

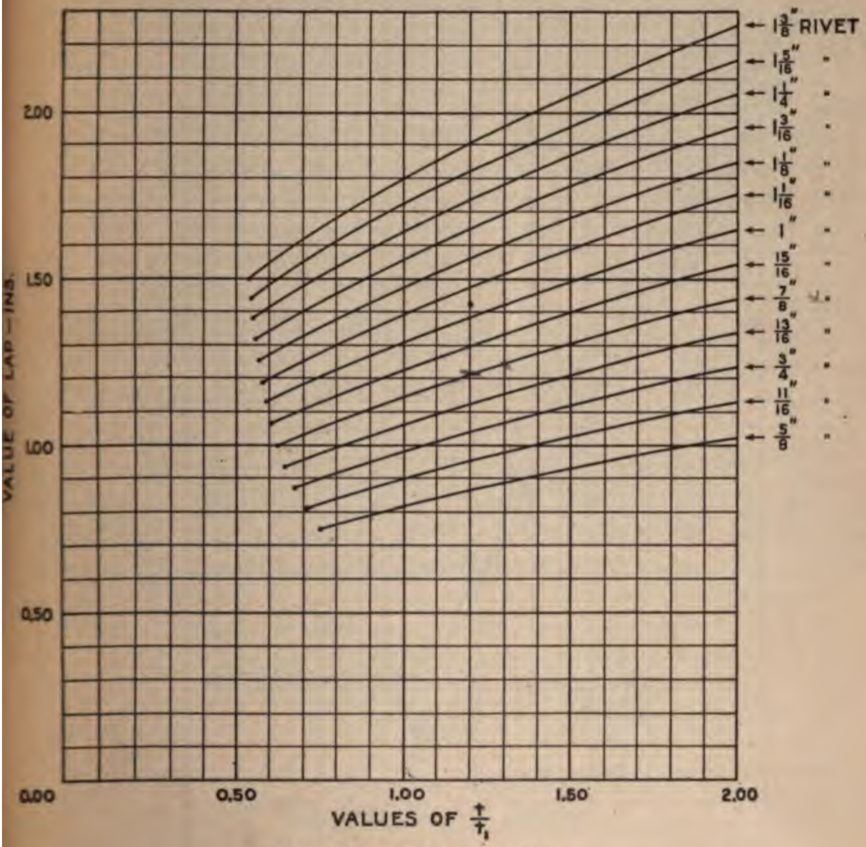
CALCULATED LAP VALUES FOR STEEL RIVETS.

Double Shearing and Crushing.

f_s = 45,000 lbs. per sq. in. f_c = 96,000 lbs. per sq. in.

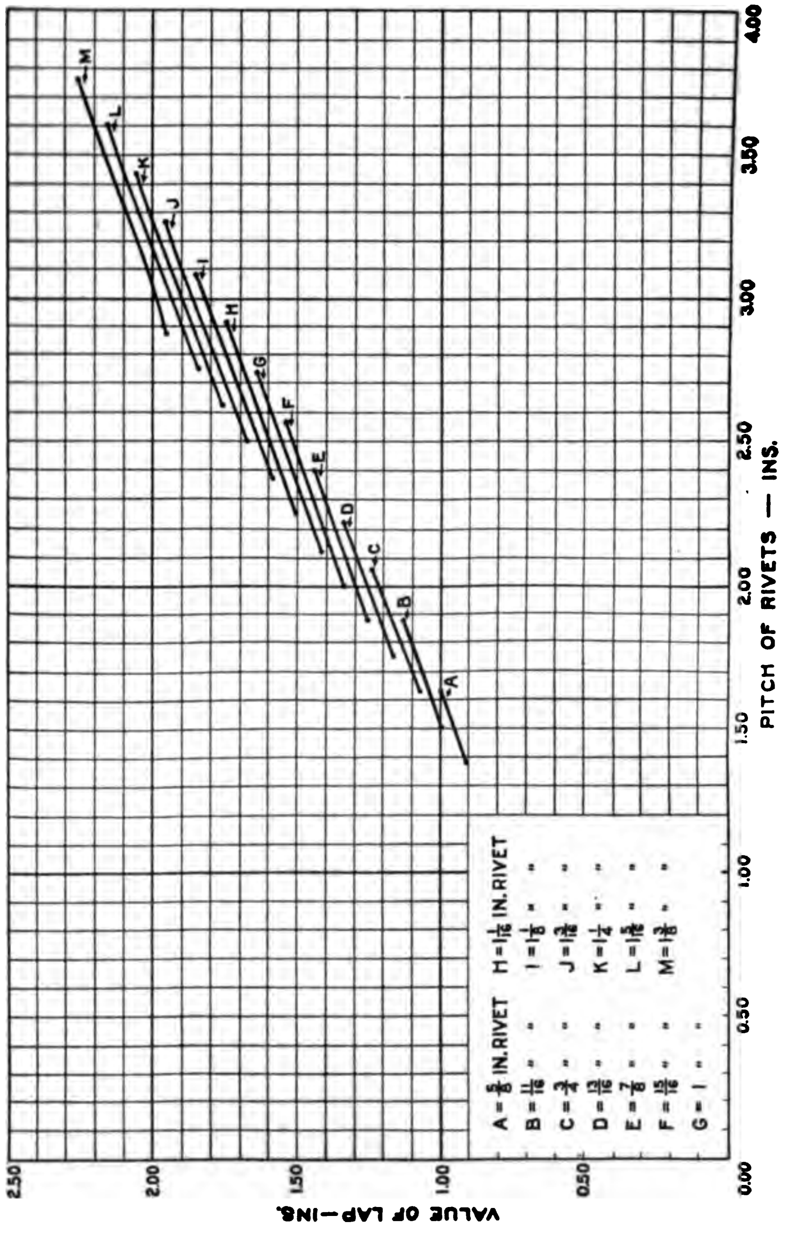
Driven rivet diam. Ins.	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	Thickness of plate forming lap. Ins.
5/8	1.03	1.03	1.03	1.03	*1.00*	*0.96*	*0.93*	*0.90*	*0.87*	*0.85*	*0.83*	*0.81*	*0.80*	**Remarks.** The following style of figures indicates failure by double shearing. *1 2 3 4 5 6 7 8 9 0* The following style of figures indicates failure by crushing. 1 2 3 4 5 6 7 8 9 0
11/16	1.13	1.13	1.13	1.13	1.13	*1.09*	*1.05*	*1.02*	*0.99*	*0.96*	*0.94*	*0.92*	*0.90*	
3/4	1.23	1.23	1.23	1.23	1.23	*1.23*	*1.18*	*1.14*	*1.11*	*1.08*	*1.06*	*1.03*	*1.01*	
13/16	1.34	1.34	1.34	1.34	1.34	1.34	*1.32*	*1.27*	*1.24*	*1.20*	*1.18*	*1.15*	*1.13*	
7/8	1.44	1.44	1.44	1.44	1.44	1.44	1.44	*1.41*	*1.37*	*1.33*	*1.30*	*1.27*	*1.24*	
15/16	1.54	1.54	1.54	1.54	1.54	1.54	1.54	1.54	*1.50*	*1.46*	*1.42*	*1.39*	*1.36*	
1	1.65	1.65	1.65	1.65	1.65	1.65	1.65	1.65	*1.63*	*1.59*	*1.55*	*1.51*	*1.48*	
1 1/16	1.75	1.75	1.75	1.75	1.75	1.75	1.75	1.75	1.75	*1.72*	*1.68*	*1.64*	*1.61*	
1 1/8	1.85	1.85	1.85	1.85	1.85	1.85	1.85	1.85	1.85	1.85	*1.81*	*1.77*	*1.73*	
1 3/16	1.95	1.95	1.95	1.95	1.95	1.95	1.95	1.95	1.95	1.95	*1.95*	*1.91*	*1.87*	
1 1/4	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	*2.04*	*2.00*	
1 5/16	2.16	2.16	2.16	2.16	2.16	2.16	2.16	2.16	2.16	2.16	2.16	2.16	*2.13*	
1 3/8	2.26	2.26	2.26	2.26	2.26	2.26	2.26	2.26	2.26	2.26	2.26	2.26	2.26	

Formula (106) for tearing laps is illustrated by the plot of Fig. 77. It must be borne in mind that this case is applicable to reduced pitches only. As soon as the pitch attains its normal value, the lap calculation reverts to Case I or III. The lines plotted in Fig. 77 are applicable, therefore, through a very short range only. At their left ends they indicate minimum laps or those $d + \frac{1}{8}$ in. in amount. The lines are continued to the right until the corresponding lap values pass over to those of Case I or III.

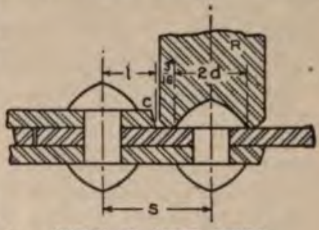


CALCULATED VALUE
OF
HALF CRUSHING LAPS

Fig. 76.


A = 5/8 IN. RIVET
B = 11/16 " "
C = 3/4 " "
D = 13/16 " "
E = 7/8 " "
F = 15/16 " "
G = 1 " "
H = 1 1/16 IN. RIVET
I = 1 1/8 " "
J = 1 3/16 " "
K = 1 1/4 " "
L = 1 5/16 " "
M = 1 3/8 " "
A
B
C
D
E
F
G
H
I
J
K
L
M
0.00
0.50
1.00
1.50
2.00
2.50
3.00
3.50
4.00
PITCH OF RIVETS — INS.
0.00
0.50
1.00
1.50
2.00
2.50
VALUE OF LAP—INS.

9. **Distance between Rows of Rivets.** — The transverse ensions of a riveted joint are determined from the calculated values explained above and the space between staggered rows, nbined with the necessities of the riveting machine. The uble-riveted butt-joint shown in tion in Fig. 78 contains three cal- lated laps, namely: at outside and ide cover plates and main plate. e width of the joint is, therefore, termined as soon as the possible ximity of the outermost rivets to e outer cover plate is known. In g 78 the die of the riveter is shown the position of driving the outer vet. Often S is made twice the lap. A safe rule in general is allow for rivet heads twice the driven diameter of the rivet and cure $\frac{3}{16}$ in. between the rivet head and lap edge. Then

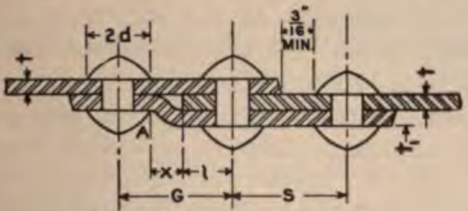


WIDTH OF BUTT JOINTS

FIG. 78.

$$S = Lap + \tfrac{3}{16}\ in. + Rivet\ diameter.$$

hen there are two inner rows of rivets in staggered order the istance between them is determined as explained in Fig. 52, age 101. If the horizontal pitch is greater than 4 ins. the full 0 per cent excess on slope lines need not be secured.



WIDTH OF WELT JOINTS

FIG. 79.

In welt joints, Fig. 79, the distance between rows depends upon he welting or bending of the cover plate in addition to the above ctors. The horizontal distance x necessary to get down from ne level of plate to the other is taken as twice the plate thickness ver which the bend is made.

Then $G = Lap + 2t + Rivet\ diameter,$

nd as before

$$S = Lap + \tfrac{3}{16}\ in. + Rivet\ diameter.$$

For symmetry S is generally made equal to G, the latter being the larger. In addition to the above requirements, the possibility of calking the lap edges must always be kept in view. If the outer rivets in Figs. 78 and 79 are placed as closely as possible to the lap, it is sometimes impossible to hold the calking tool at the correct angle. Room must be provided for holding the latter at about 30° with the plate below.

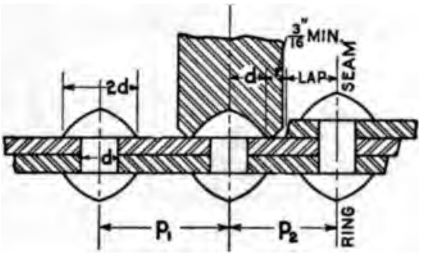
50. Insertion of Joints. — Having computed the pitch and laps, and determined the transverse dimensions of the joint, it remains to space the pitch properly between ring seams. On account of the reduced strength and ductility of boiler plate when submitted to tension at right angles to the direction of rolling, pressure vessels are seldom made up with the hoop tension crosswise of the fibre. Most boilers and tanks contain three or more cylindrical courses, placed inside and outside in alternation. The width of each course is determined by the exigencies of the setting, the width of plate readily procurable and the reach of the riveting machine. It is well in multi-course vessels to have two or more of the joints in duplicate for ease in laying out. No attempt is made to keep the pitch in integral sixteenths of an inch. With the distance between ring seams expressed in binary fractions, the pitch is spaced to fit. Small changes of 2 or 3 per cent in the pitch are permitted without considering that the former calculations have been affected. All the other dimensions of riveted work, aside from the pitch, should be expressed in binary fractions not smaller than one-sixteenth inch.

The same size of rivets should be used in both ring and longitudinal seams in order to make as little disturbance as possible where they cross one another. Longitudinal joints are never placed in line and thus the confusion of multiple plate thicknesses and complex rivet arrangements is avoided. Whatever the irregularities necessarily introduced where the ring and longitudinal seams cross, the safe calking pitch must never be exceeded and the joint efficiency must not be unduly lowered.

While very few longitudinal butt-joints have given way to absolute failure, local deterioration and distortion have often taken place due to careless rivet arrangement at the ring seam juncture. Great care must be exercised, therefore, to avoid an undue expanse of unstayed plate as well as a lowered efficiency when seeking the necessary room for driving rivets in this region.

When a single-riveted ring seam crosses a longitudinal joint if possible a point should be selected for the juncture so as to leave the outside or long pitch undivided. The outer row is already the weakest place in the joint and if more rivet holes are placed on that line, the efficiency falls rapidly. For instance, the efficiency of a triple-riveted butt-joint using rivets one inch in diameter is reduced from 89 to 78 per cent by the introduction of one more rivet hole on the outside row. The outside pitch should, therefore, be left whole or nearly so, where the longitudinal and transverse joints meet.

In chain-riveted seams, the position of the last rivet in the joint next to the ring seam must always be such as to secure room for



SEAM JUNCTURES

Fig. 80.

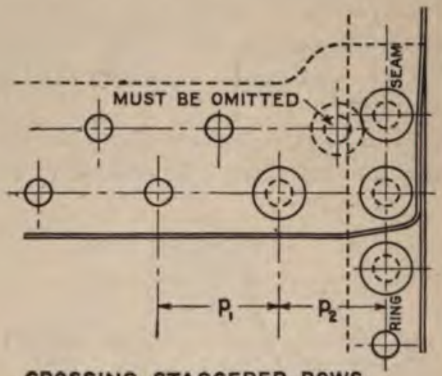
the die of the riveter to act, Fig. 80. If the natural pitch of the longitudinal joint is such that

$$p_2 \geqq \textit{Rivet diam.} + \tfrac{3}{16}\ \textit{in.} + \textit{Ring seam lap},$$

then p_2 may be kept equal to p_1, and there is no need of changing the ordinary pitch of the joint. When p_2, however, is naturally too small, the last pitch of each joint must arbitrarily be made greater to accommodate the driving of the rivet immediately adjacent to the ring seam.

The crossing of the lengthwise joint by the crosswise one is generally accompanied with some difficulty when the former contains rivets in staggered order. With usual rivet sizes and thicknesses of plate it is not possible to cross a staggered row without the omission of at least one of its rivets.

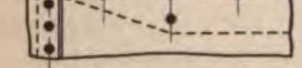
In Fig. 81 there is not generally enough room to drive the last rivet on the rear row because of its proximity to the ring seam lap. Hence this rivet must be omitted. To compensate for its loss, the last pitch p_2 is shortened to about 0.8 its original amount. This lowers the efficiency to a slight degree. The rivet thus dropped must not be one upon the calked edge, since the pitch on that line is supposedly as large as it can be without leakage.



CROSSING STAGGERED ROWS

Fig. 81.

The ends of the outside cover plates of inner courses, when of moderate thickness, are thinned to a sharp edge and tucked under the lap of the next outer course. The sides of such cover plates are also thinned near the ends so as to facilitate tight calking. The ends of inside cover plates are usually cut off just short of the ring seam. In very staunch work they may extend into the ring seam and participate in the riveting.



BUTT JOINT ENDS

Fig. 82.

With very thick plates it is so difficult to tuck covers under successive courses that the method shown in Fig. 82 has been employed by the International Engineering Co. of South Framingham, Mass., in heavy boilers. The pitch along the calked edge of the staggered row is kept uniform. Instead of dropping an inner rivet, it is displaced a little to the right in order to provide room for the soft steel pipe plug e. Then the cover is rounded at b and cut short as shown. This method permits easy calking of the ring seam, pipe plug and cover plate. With large pitches and thick plates it is a satisfactory method.

The following riveted-joint illustrations, Figs. 83 to 98 inclusive, show the practical arrangement of rivets at junctures for a variety of seams. The general conventions adopted in the cuts are as follows:

A shows the joint end of an inside course underlapping the next one to the left.

B indicates the joint end of an external course overlapping the next one to the right.

C is a section on the ring seam *MN* at the right, showing the thinning necessary to insure a smooth surface for the next inner course to rest against.

d is the driven rivet diameter.

E is a section of the run of joint.

F shows the manner in which a joint is made up where its end occurs at an uptake or similar opening. The overlapping plate is calked on the outside to the extreme edge *Q* and on the underside back to the tube sheet lap.

G is a section on the ring seam *XY* at the left, showing the thinning necessary to insure a smooth surface for the next outer course to rest against.

HK is the horizontal line on which the section of the tube sheet and joint is taken.

MN is the vertical line on which the ring seam section at the right is taken.

p is the ordinary pitch of the rivets.

p_1 is the divided pitch in joints where the outside and inside spacing is not the same.

p_2 is the end pitch, which may be enlarged or reduced, depending upon the type of joint.

Q is the point to which the joint is calked for tightness in an endwise direction. In lap joints the plate edges are calked. In butt-joints a chip or wedge is driven between the abutting plates. Sometimes point *Q* comes well up in the uptake opening and again at the extreme corner of it.

XY is the vertical line on which the ring seam section at the left is taken.

Fig. 83. Provided the ordinary pitch p gives sufficient room to drive the rivet next to the ring seam, p_2 is made equal to p, and the pitch is merely spaced between ring seams. The rivet next to the tube sheet flange is gouged slightly to permit assembling.

Fig. 84. The last pitch p_2 is shortened to about 0.8 of its original value to compensate for the rivet dropped on the rear row. The ordinary pitch p is then spaced in regular order.

Fig. 85. The rivet rows are placed at a suitable distance from one another to give the usual diagonal pitch. A rivet on the middle row is dropped at the ring seam and p_2 consequently shortened to 0.8 p.

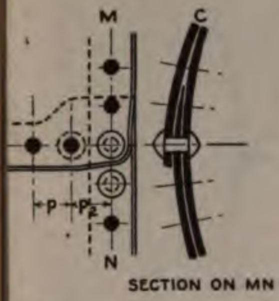
Fig. 86. If the ordinary pitch p is sufficiently large to accommodate the ring seam lap, the spacing is uniform from one ring seam to the other. If not, p_2 is made greater than p by the required amount.

Fig. 87. After having subtracted twice the value of p_2 from the seam length, the pitch p is spaced in uniformly.

Fig. 88. The transverse distance between rows of rivets is made sufficient to insure failure along longitudinal lines as shown in Fig. 52, page 101. The outer pitch p is divided into two inside pitches p_1. After having subtracted twice the value of p_2 from the seam length, the inside pitch p_1 is spaced in uniformly so that the total number of such pitches will be some such number as 17 or 19, a multiple of two, plus one.

Fig. 89. In joints using a fraction of the outer pitch on the inner row, it is best to space the divided pitch. If p_1 will give room for the ring seam lap, the small pitch should be spaced uniformly from end to end of the joint. The seam should contain an even number of small pitches.

Fig. 90. The alternative arrangement of the single-riveted welt-plate joint calls for a grouping of three small pitches with two large pitches. Therefore, if p_1 will permit the driving of the rivet next to the ring seam, the small pitch should be spaced uniformly throughout the joint. The total number of small pitches should then be a multiple of three. On the other hand, if the last space must be made larger than p_1, twice p_2 should be subtracted from the distance between ring seams and the remaining distance divided into small pitches the total number of which shall be some such number as 22 or 25, a multiple of three, plus one.



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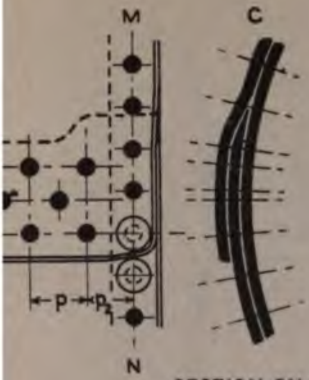

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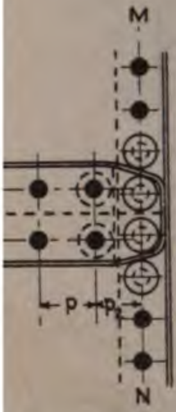

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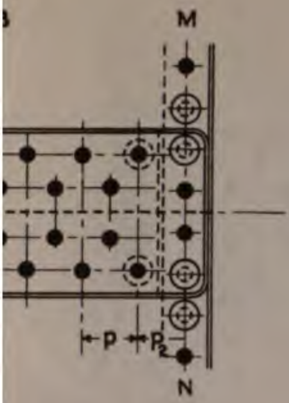

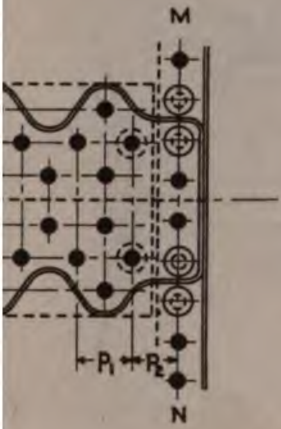

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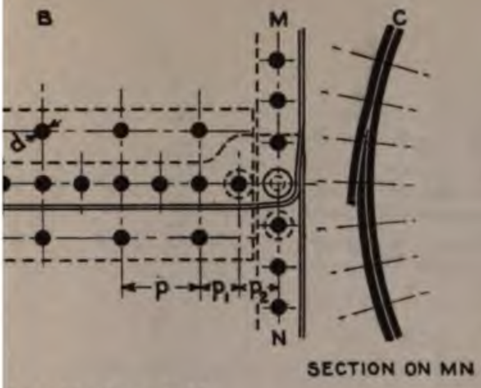

NEXT TO RING SEAM

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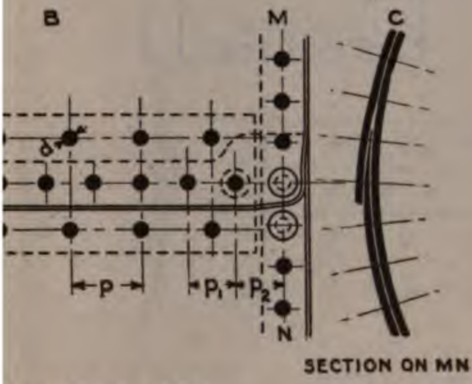

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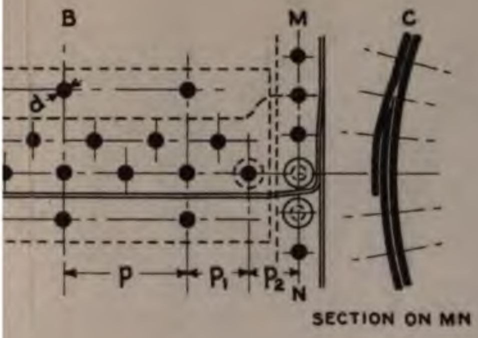

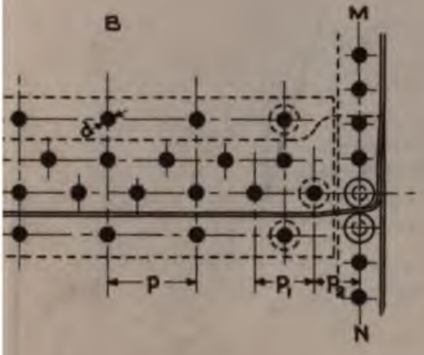

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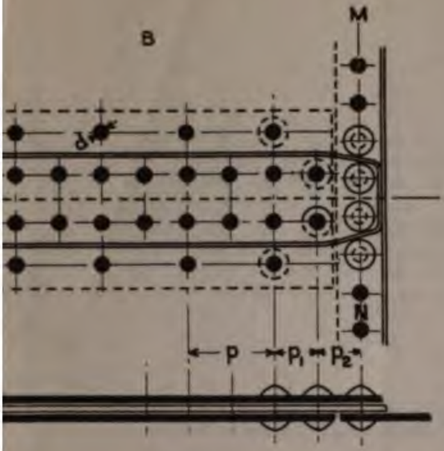

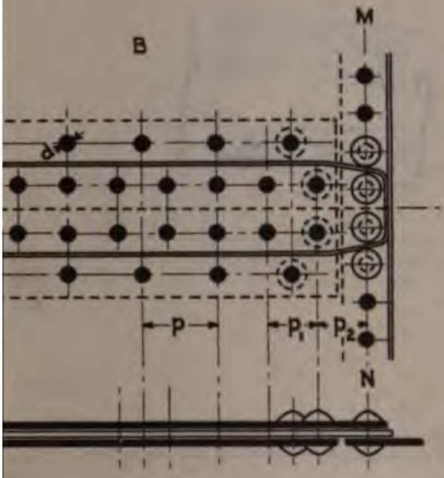

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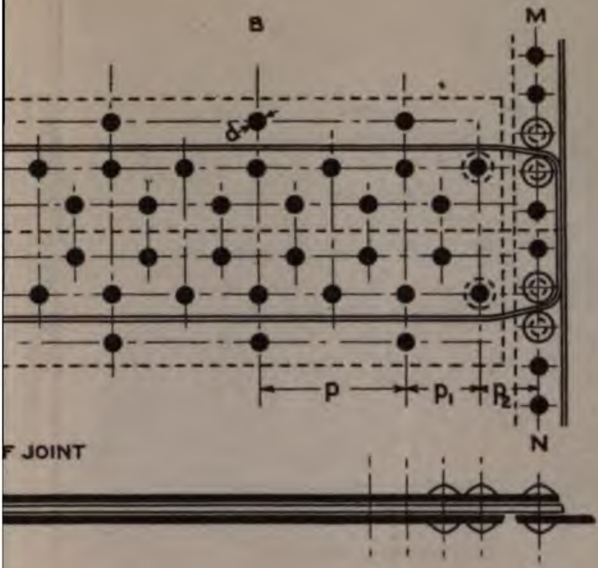

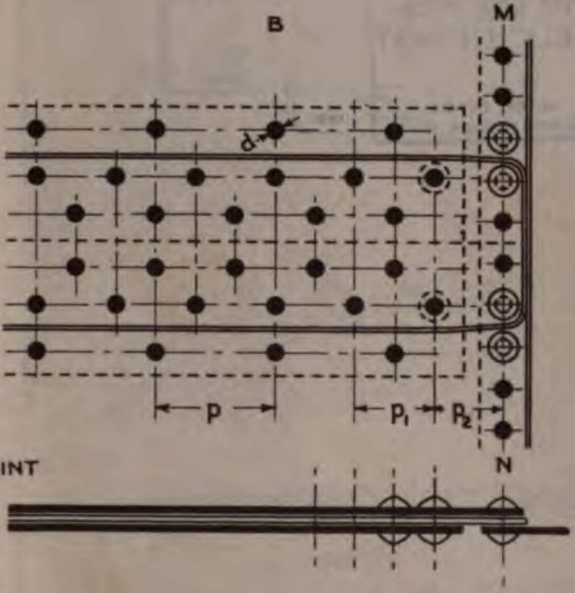

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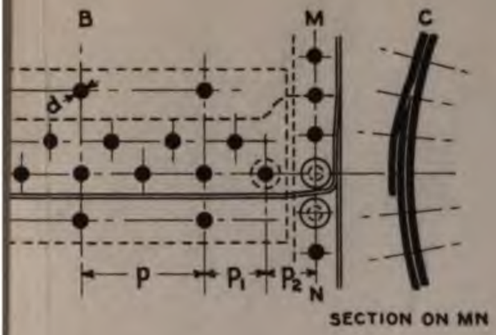

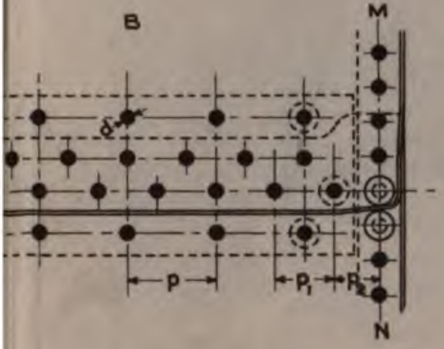

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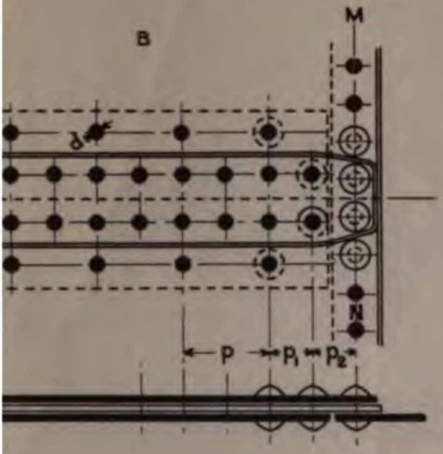

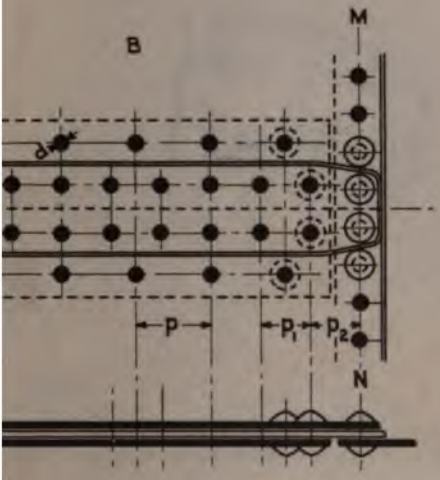

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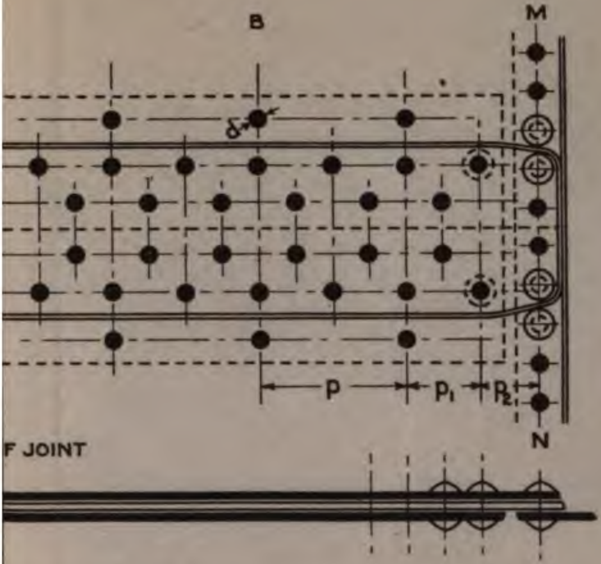

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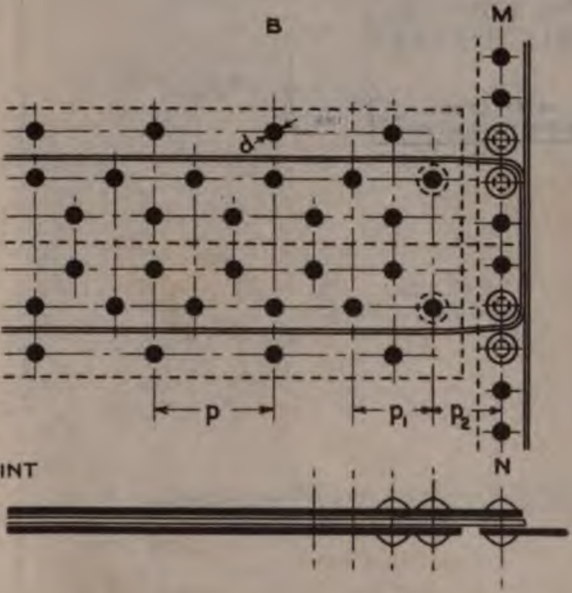

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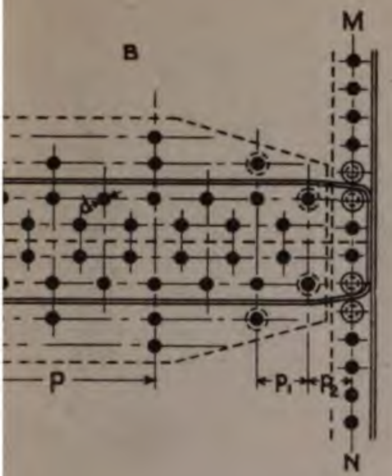

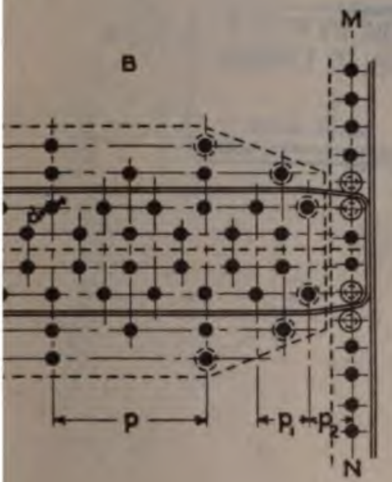

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Fig. 91. The staggered riveting in this joint calls for the usual reduced pitches at the ends. The total number of inside pitches must be a multiple of two, in order to avoid unwarranted rivet holes on the outer row.

Fig. 92. After having determined p_2 and subtracted twice its value from the seam length, the inside pitch p_1 should be arranged in groups of three, plus one. The outer pitches will then be symmetrically arranged throughout the joint.

Fig. 93. Provided the end pitches will give room for driving the rivets adjacent to the ring seam, the small pitch p_1 should be spaced in groups of two throughout the seam.

Fig. 94. If p_1 is large enough to secure room for the end riveting, the joint may be uniformly spaced throughout, in multiples of three small pitches. If p_2 has to be made arbitrarily larger than p_1, the total number of inside pitches will be a multiple of three, plus one.

Fig. 95. With twice the values of p_2 subtracted from the joint length, the small pitch p_1 should be uniformly spaced throughout in multiples of two.

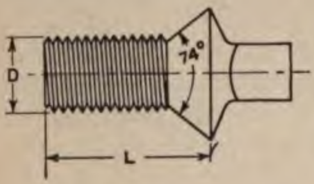
Fig. 96. To insure symmetry the seam length, lessened by twice the end spaces p_2, should contain a total number of pitches p_1 consisting of a multiple of three, plus one.

Fig. 97. In quadruple-riveted butt-joints there are various possible methods of insertion. If the large pitch is to be kept intact and approximately the full efficiency secured at the ends of the joint, the total number of small pitches p_1, after having deducted p_2, should be a multiple of four, plus two. If this is especially difficult to obtain, the grouping may be simply in twos, there being, consequently, a small theoretical loss in efficiency.

Fig. 98. The end spacings p_2 having been deducted from the joint length, the small pitch should be arranged to comprise a number consisting of a multiple of three, plus one. If this appears to be impossible, the joint can be suitably arranged, there being a slight loss in efficiency, with inside pitches simply in groups of three.

Some designers object to the location of rivets in outer rows circumferentially opposite those upon inner rows, claiming that the holding power of rivets so placed is impaired. While it is doubtful whether or not there is any basis for such a belief, joints may be inserted without rivets opposite one another, provided fractional pitches are used at the ends. The principles explained above are fully applicable to rivet arrangements of this character.

51. Patches. — There are two classes of patches, namely, hard and soft, applied to pressure vessels to stop local deterioration. When cracks or bulged plates appear, the metal affected must be entirely removed before the patch can be applied. The hard patch consists of a piece of steel plate, accurately shaped, and permanently riveted to place. Soft patches are often applied temporarily or in case of emergency and are not supposed to constitute a permanent repair. The metal used for the latter is generally soft wrought iron which can be shaped to fit the required place by hammering cold. Soft patches are usually set in boilermaker's putty or with lead gaskets to prevent leakage and are secured in place by threaded patch bolts. Fig. 99 shows a typical patch bolt. Table V gives their usual proportions. They are introduced from the side toward the fire and when screwed home securely the protrusion, by which they are driven, is chipped off.



PATCH BOLT

Fig. 99.

TABLE V.

PROPORTIONS OF BOILER PATCH BOLTS.

12 threads per inch.

Diam. *D*. In.	Usual lengths, *L*. Ins.
$\frac{1}{2}$	$\frac{3}{4}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$
$\frac{5}{8}$	$\frac{3}{4}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$
$\frac{3}{4}$	.. 1, $1\frac{1}{4}$, $1\frac{1}{2}$
$\frac{7}{8}$	.. 1, $1\frac{1}{4}$, $1\frac{1}{2}$
1	.. .. $1\frac{1}{4}$, $1\frac{1}{2}$

When the form of the patch is very complex, the following method may often be used to gain an accurate idea of its shape. A hole, drilled in the center of the locality which the patch is to cover, will permit the introduction of a quantity of plastic plaster of Paris. After compacting this against the injured plate and allowing it to set, the cast thus obtained may be smoothed up and used as a pattern in the foundry. The anvil formed in this manner may be used for guidance in shaping the patch so as to secure a good fit.

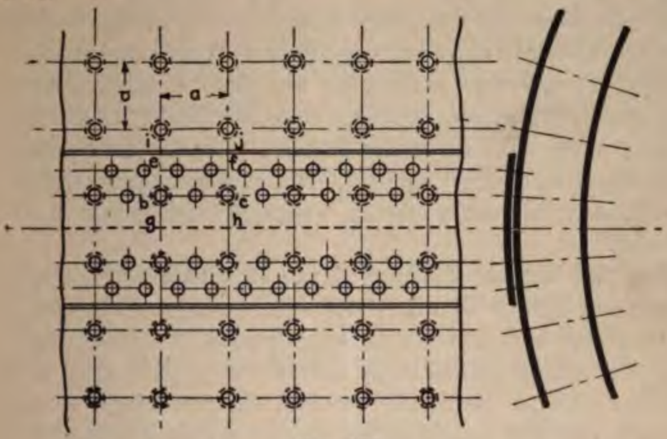


Fig. 100.

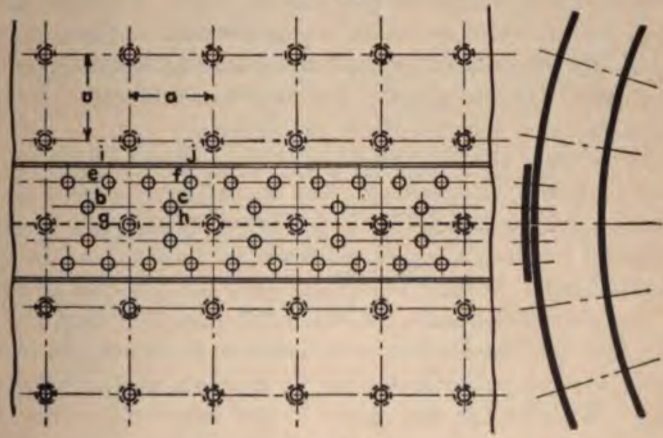


Fig. 101.

WATER LEG JOINTS

52. Water Leg Joints. — The spacing of stay bolts in the water legs of locomotive and vertical boilers often introduces considerable complexity when riveted joints must be accommodated in these members as well. Since the circulation space is very valuable in this region, inside cover plates are generally omitted and the joints shown in Figs. 100 and 101 are used instead.

These joints consist essentially of special double-riveted lap joints in duplicate, and were it not for the holding power of the stay bolts, considerable distortion would probably result. In the actual case, however, the water channel is held rigidly to shape by the stays and no trouble is experienced in this direction. Two methods are shown of making up the joint without disturbance to the stay bolt spacing. Since, due to the rigidity of the staying, the hoop tension in the outer furnace sheet does not approach the expected theoretical amount, the low efficiency of the joint is not critical. The procedure in designing such joints is at best empirical.

When the natural spacing of the stay bolt will permit, the arrangement shown in Fig. 101 should be adopted. The rivets in these joints should be the same in diameter as those used in the other longitudinal seams. The outside cover plate may be made somewhat thinner than the external furnace sheet to provide flexibility, but care should be taken that the longitudinal pitch of the rivets is well within the safe calking limit. The laps and distance between rivet rows should be calculated in the usual manner.

When the width of cover plate determined above is so great as to interfere with the placing and heading of the stay bolts, the method shown in Fig. 100 must be followed.

In either form, the laps should be at least equal to those required by the rivet size; the calking pitch should never exceed the safe amounts given in Table XXIII, and the diagonal pitch should be determined by the principles set down in the previous discussion of that subject. Further than this, these joints do not lend themselves to exact calculation. Great care must be taken whenever such joints run beyond the stay-bolted area, to change them at once to typical butt-joints with both inside and outside cover plates. Without the assistance of the stay bolts, these joints would not be strong enough to hold the hoop tension of a cylindrical shell even under low pressures.

53. Experimental Results. — The question is often asked as to how near the foregoing theory of riveted-joint design stands to the results of actual tests. While experiments upon riveted joints are generally unsatisfactory because of the absence of heat and corrosion, nevertheless many such tests have been made, and the results afford an interesting comparison with theoretical calculations. In 1887 an elaborate series of tests was carried out at the U. S. Arsenal, Watertown, Mass., upon joints supplied by the Bureau of Steam Engineering of the U. S. Navy. The joint specimens were all of the butt-type with double cover plates. Some of the joints used cover plates of equal width and others of the extended type. All the joints were tested to destruction, minute observations being made upon the behavior of the rivets and plate. The efficiency was calculated by dividing the maximum load at which the joint failed by the tensile strength of the solid plate as determined from previous experiments. To make this quantity entirely comparable with theoretical efficiencies, a repeating section of the joint should have been used in the experiments. This was not done owing to the difficulty of driving rivets along the edge. Hence the practical results do not express the exact efficiency as heretofore defined. The joints were so wide, however, that the small margin outside of the repeating section probably had an inconsiderable effect upon the efficiency. The results of all the tests in this series are compared with the theoretical efficiencies in Table VI. While not of vital importance the comparison

TABLE VI.
EXPERIMENTAL EFFICIENCY OF RIVETED JOINTS.

Tested at the U. S. Arsenal, Watertown, Mass.

No. of test.	Type of joint.	Experimental efficiency. Per cent.	Calculated efficiency. Per cent.
910	Double-riveted butt	79.8	74.0
911	Double-riveted butt	76.2	
912	Double-riveted butt	79.5	
913	Triple-riveted butt	80.9	78.1
914	Triple-riveted butt	75.2	
915	Triple-riveted butt	82.5	
916	Double-riveted butt	68.7	71.7
917	Double-riveted butt	70.7	
918	Double-riveted butt	69.9	
919	Triple-riveted butt	74.1	76.7
920	Triple-riveted butt	75.0	
921	Triple-riveted butt	74.1	
922	Quadruple-riveted butt	79.7	76.2
923	Quadruple-riveted butt	81.5	
924	Quadruple-riveted butt	78.4	
925	Quadruple-riveted butt	79.7	82.6
926	Quadruple-riveted butt	84.3	
927	Quadruple-riveted butt	85.8	
928	Quadruple-riveted butt	80.6	91.3
929	Quadruple-riveted butt	77.8	
930	Quadruple-riveted butt	77.7	
931	Triple-riveted butt	81.2	78.1
932	Triple-riveted butt	80.5	
933	Triple-riveted butt	80.1	

is interesting. In some of the joints, especially Nos. 928, 929 and 930, the rivet rows were placed so close together that at rupture the tearing of the plate occurred along diagonal lines much less in length than that of the usual longitudinal ones. Hence the efficiency was low.

54. Riveting around Circular Openings. — Figs. 102 and 103 show respectively single- and double-riveted seams around manhole openings. Such seams are either elliptical or of an approximately elliptical shape made up of semi-circular arcs and flat sides. If of true elliptical shape they are laid out by template on the flat plate before rolling. If of approximate form they are struck on the flat plate with compasses and straight edge.

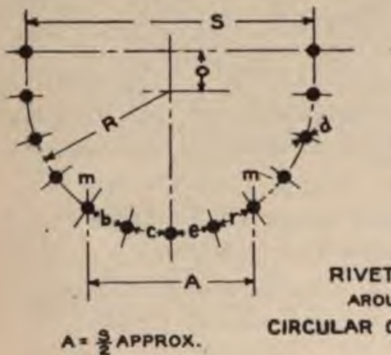


Fig. 102.

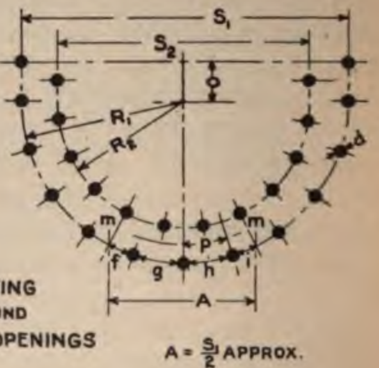


Fig. 103.

The positions of the rivets on these openings are always such as insure a symmetrical arrangement, thus making the number even. Openings of this shape are correctly placed with their long dimension in a circumferential direction with regard to the boiler, thus removing as little plate in a longitudinal direction as possible. The end tension in a cylinder is never more than one-half the hoop tension, hence this is a proper arrangement of an elliptical opening. It is always best to investigate any special feature of this kind as to the actual factor of safety. The joint, Fig. 102, close to its central portion *ce*, is in the most critical condition and at this point would probably have a low factor of safety. It is believed that in such joints, a length A about $\frac{1}{2}$ S for single-riveted rings of ordinary size (not over 16 ins. opening) may be taken and the assumption made that the stiffness of the plate will cause the more favorably situated spaces b and r to aid the weaker spaces c and e. The limiting lines mm for the sections considered should be so located as to pass through either the center of a rivet or the center of a space, and at the same time leave the distance A approximately as shown. This secures a portion of the joint similar to the repeating section chosen in other joints. Inasmuch as the tearing of the plate is here confined to chordal lines between rivets, while the hoop tension varies with the length of the element of the cylinder, these joints are more efficient than straight joints of the same pitch, especially when a width of four or more pitches is considered. Since the calculated pitch can rarely be made to fit the desired pitch line without change, it is necessary to calculate all the possible resistances to failure of both plate and rivets.

Proceeding as in the general case of riveted joints:

I. Tearing.

$$Resistance = (b + c + e + r)\,tf_t \quad . \quad . \quad . \quad (107)$$

II. Shearing.

$$Resistance = 4\left(\frac{\pi d^2}{4}\right)f_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (108)$$

III. Crushing.

$$Resistance = 4\,dtf_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (109)$$

With P the working pressure and R the radius of the cylindrical shell,

$$IV.\ Hoop\ tension\ for\ length\ A = PRA. \quad . \quad . \quad . \quad . \quad . \quad . \quad (110)$$

Then
$$Factor\ of\ Safety = \frac{Least\ Resistance}{Hoop\ Tension}. \quad . \quad . \quad . \quad (111)$$

For the double-riveted ring, Fig. 103, the pitch on the inner or calked row should be uniform to insure tightness. This may be secured by laying out one pitch of the theoretical amount as shown, on the circle half-way between the rivet circles. A radial line, drawn to the center of the semi-circular pitch lines, indicates graphically the desired size of the pitch on the inner and outer rows. If calking takes place on the inner circle, the inner pitch should be spaced evenly and the outer rivets located radially opposite the centers of the inner spaces. Should calking occur on the outer edge, the outer spaces are made uniform, thus always leaving the pitch uniform on the calked edge.

The resistances to failure are then:

I. Tearing.

$$\text{Resistance} = (f + g + h + i)\,tf_t \quad . \quad . \quad . \quad (112)$$

II. Shearing.

$$\text{Resistance} = 7\left(\frac{\pi d^2}{4}\right)f_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (113)$$

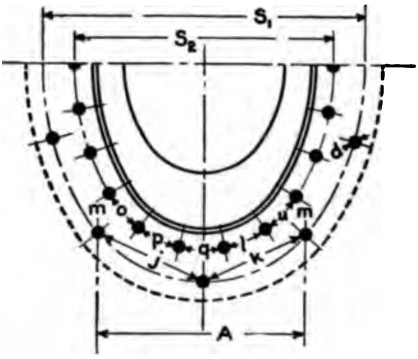
III. Crushing.

$$\text{Resistance} = 7\,dtf_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (114)$$

As before

IV. Hoop tension for length $A = PRA$. . . . . . (115)

And
$$\text{Factor of Safety} = \frac{\text{Least Resistance}}{\text{Hoop Tension}} \quad . \quad . \quad . \quad . \quad (116)$$



SPECIAL MANHOLE SEAM

$A = \frac{S_1}{2}$ APPROX.

Fig. 104.

Fig. 104 shows the elliptical manhole seam recommended by the Massachusetts Boiler Rules. Alternate rivets are omitted in the outer row. Proceeding as before, the resistances to failure are:

I. Tearing along outer row.

$$\text{Resistance} = (j + k)\,tf_t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (117)$$

II. Tearing along inner row and shearing rivets.

$$\text{Resistance} = (o + p + q + l + u)\,tf_t + 2\left(\frac{\pi d^2}{4}\right)f_s \quad (118)$$

III. Tearing along inner row and crushing rivets.

$$\text{Resistance} = (o + p + q + l + u)\,tf_t + 2\,dtf_c \quad . \quad . \quad (119)$$

IV. Shearing all rivets.

$$\textit{Resistance} = 7\left(\frac{\pi d^2}{4}\right)f_s \quad \ldots\ldots\ldots \quad (120)$$

V. Crushing all rivets.

$$\textit{Resistance} = 7\,dtf_c \quad \ldots\ldots\ldots \quad (121)$$

VI. Hoop tension for length $A = PRA$. . . . . . (122)

Then $$\textit{Factor of Safety} = \frac{\textit{Least Resistance}}{\textit{Hoop Tension}} \quad \ldots \quad (123)$$

An application of the above method to the seam of the manhole in the Horizontal Return Tubular Boiler of Chap. V results in a factor of safety of 5.40. Such curved seams partake, to a certain degree, of the characteristics of helical seams and as such are probably stronger than the above approximate investigation would indicate.

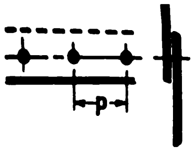
55. Tabulations. — In order to facilitate the use of the foregoing theory of riveted joints in practical work, the following tables of maximum pitches and efficiencies and maximum calking distances have been appended.

Tables VII to XXII inclusive give the maximum pitch to be used in a wide variety of joints with the several rivet diameters and thicknesses of plate. Just below the pitch value in each case is specified the corresponding joint efficiency. These are to be regarded as the maximum values consistent with well designed joints. When reduced values are arbitrarily used for the pitch, the efficiency is correspondingly lowered. The spaces left blank in the pitch tables indicate in certain cases that rivet heads 2 d in diameter will come less than $\frac{1}{8}$ in. apart, either in straight or diagonal lines, and hence cannot be driven. In other cases the omission of the pitch value signifies that the rivet in the table is less than the minimum allowable size for the plate in question, and hence could not be used in a properly designed joint. The manner of failure in each pitch calculation is indicated by the style of figures in which the results are printed. An explanation of the significance of the styles of figures is placed under the column headed "Remarks" in each table. The inside cover plate thickness given is that recommended by the Massachusetts Boiler Rules.

Table XXIII of calking pitches indicates the maximum values consistent with tightness at a variety of working pressures. The thickness of plate specified is always that of the one which is calked, in general the outside cover plate.

In order to secure a fair distribution of stress throughout the joint the thicknesses of the butt-straps or cover plates must be sensibly equal. To prevent tearing of the covers along the center line of the joint the thickness of each must exceed by a reasonable margin half that of the main plate. In the case of the outer cover this margin is added to enhance its stiffness and enable it the more effectively to resist leakage of the confined fluid. When fulfilling the above conditions the outside cover plate does not enter into the pitch calculation. Hence if necessary its thickness may be increased to meet the necessities of a given problem.

The function of the inside cover, aside from its office as a portion of a riveted fastening, is to protect the main plate from corrosion. It must therefore have a thickness far in excess of the theoretical requirements discussed in Art. 42, page 124. In butt and welt joints with extended inside cover plates, so long as the outer rows of rivets fail by shearing, the thickness of the inside cover does not affect the pitch calculation and may consequently be made greater than the amount given in the tables if the expected corrosion warrants it. With crushing rivets however the pitch value is dependent upon the bearing power of the inside cover. Hence in the latter case the stipulated thickness alone should be used.



Joint A.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Single-riveted Lap Joint. — Joint A.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Thickness of main plate. Ins.
$\frac{5}{8}$	*1.63* *61.6*	*1.43* *56.3*	....	....	....	....	....	....	....	....	....	....	....	Remarks. The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question: 1 2 3 4 5 6 7 8 9 0 The following style of figures indicates failure by single shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable: *1 2 3 4 5 6 7 8 9 0*
$\frac{11}{16}$	1.89 63 6	*1.66* *58.6*	*1.50* *54.1*	....	....	....	....	....	....	....	....	....	....	
$\frac{3}{4}$	2.06 63.6	*1.91* *60.7*	*1.71* *56.2*	....	....	....	....	....	....	....	....	....	....	
$\frac{13}{16}$	2.22 63.6	*2.17* *62.5*	*1.94* *58.2*	*1.78* *54.4*	....	....	....	....	....	....	....	....	....	
$\frac{7}{8}$	2.40 63.6	2.40 63.6	*2.19* *60.0*	*2.00* *56.2*	....	....	....	....	....	....	....	....	....	
$\frac{15}{16}$	2.57 63.6	2.57 63.6	*2.44* *61.6*	*2.23* *57.9*	*2.07* *54.6*	....	....	....	....	....	....	....	....	
1	2.75 63.6	2.75 63.6	*2.71* *63.1*	*2.47* *59.5*	*2.29* *56.2*	*2.14* *53.3*	....	....	....	....	....	....	....	
$1\frac{1}{16}$	2.92 63.6	2.92 63.6	2.92 63.6	*2.72* *60.9*	*2.51* *57.7*	*2.35* *54.8*	....	....	....	....	....	....	....	
$1\frac{1}{8}$	3.09 63.6	3.09 63 6	3.09 63.6	*2.98* *62.3*	*2.75* *59.1*	*2.57* *56.3*	*2.43* *53.6*	....	....	....	....	....	....	
$1\frac{3}{16}$	3.26 63.6	3.26 63.6	3.26 63.6	*3.26* *63.5*	*3.00* *60.4*	*2.80* *57.6*	*2.64* *55.0*	*2.51* *52.6*	....	....	....	....	....	
$1\frac{1}{4}$	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63 6	*3.26* *61.6*	*3.03* *58.8*	*2.86* *56.2*	*2.71* *53.9*	....	....	....	....	....	
$1\frac{5}{16}$	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	*3.53* *62.8*	*3.28* *60.0*	*3.08* *57.4*	*2.92* *55.1*	*2.79* *52.9*	....	....	....	....	
$1\frac{3}{8}$	3.78 63.6	3.78 63.6	3.78 63.6	3.78 63.6	3.78 63.6	*3.54* *61.1*	*3.32* *58.6*	*3.14* *56.2*	*3.00* *54.1*	*2.87* *52.1*	....	....	....	

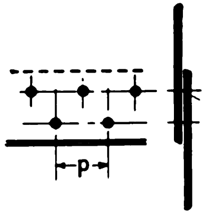


TABLE VIII.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Double-riveted Lap Joint. — Joint B.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Joint B.

Driven rivet diam. Ins.	Thickness of main plate. Ins.												
	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
$\frac{5}{8}$	*2.63* *76.3*	*2.23* *72.0*	*1.96* *68.2*	*1.77* *64.7*	....	....	....	....	....	....	....	....	....
$\frac{11}{16}$	3.09 77.7	*2.63* *73.8*	*2.31* *70.2*	*2.08* *66.9*	*1.90* *63.9*	....	....	....	....	....	....	....	....
$\frac{3}{4}$	3.37 77.7	*3.06* *75.5*	*2.68* *72.0*	*2.40* *68.8*	*2.20* *65.9*	*2.04* *63.1*	....	....	....	....	....	....	....
$\frac{13}{16}$	3.65 77.7	*3.53* *77.0*	*3.07* *73.6*	*2.75* *70.5*	*2.51* *67.6*	*2.32* *65.0*	....	....	....	....	....	....	....
$\frac{7}{8}$	3.93 77.7	3.93 77.7	*3.50* *75.0*	*3.12* *72.0*	*2.84* *69.2*	*2.63* *66.7*	*2.45* *64.3*	....	....	....	....	....	....
$\frac{15}{16}$	4.21 77.7	4.21 77.7	*3.95* *76.3*	*3.52* *73.4*	*3.20* *70.7*	*2.95* *68.2*	*2.74* *65.8*	*2.58* *63.7*	....	....	....	....	....
1	4.49 77.7	4.49 77.7	*4.43* *77.4*	*3.94* *74.6*	*3.57* *72.0*	*3.28* *69.6*	*3.06* *67.3*	*2.87* *65.1*	*2.71* *63.1*	....	....	....	....
$1\frac{1}{16}$	4.77 77.7	4.77 77.7	4.77 77.7	*4.38* *75.7*	*3.97* *73.2*	*3.64* *70.8*	*3.39* *68.6*	*3.17* *66.5*	*3.00* *64.5*	*2.85* *62.7*	....	....	....
$1\frac{1}{8}$	5.05 77.7	5.05 77.7	5.05 77.7	*4.84* *76.8*	*4.38* *74.3*	*4.02* *72.0*	*3.73* *69.8*	*3.49* *67.5*	*3.29* *65.9*	*3.13* *64.0*	*2.98* *62.3*	....	....
$1\frac{3}{16}$	5.34 77.7	5.34 77.7	5.34 77.7	*5.33* *77.7*	*4.81* *75.3*	*4.41* *73.1*	*4.09* *70.9*	*3.82* *68.9*	*3.60* *67.0*	*3.42* *65.2*	*3.26* *63.5*	*3.12* *61.9*	....
$1\frac{1}{4}$	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	*5.26* *76.3*	*4.82* *74.1*	*4.46* *72.0*	*4.17* *70.0*	*3.93* *68.2*	*3.72* *66.4*	*3.55* *64.8*	*3.39* *63.1*	....

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by single shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0

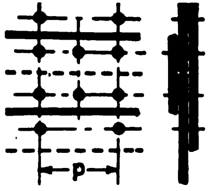

Joint K.

TABLE XVII.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Double-riveted Butt-joint. — Joint K.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	Main plate thickness. Ins.
	1/4	1/4	5/16	3/8	7/16	7/16	1/2	9/16	9/16	5/8	5/8	11/16	3/4	Inside cover plate thickness. Ins.
5/8	**3.81** **83.6**	**3.61** **82.7**	**3.48** **82.0**	....	....	....	....	....	....	....	....	....	....	
11/16	4.29 84.0	4.05 83.0	**3.90** **82.4**	**3.78** **81.8**	....	....	....	....	....	....	....	....	....	
3/4	4.68 84.0	4.42 83.0	**4.33** **82.7**	**4.19** **82.1**	....	....	....	....	....	....	....	....	....	
13/16	5.07 84.0	4.78 83.0	**4.78** **83.0**	**4.62** **82.4**	**4.50** **81.9**	....	....	....	....	....	....	....	....	
7/8	5.46 84.0	5.15 83.0	5.20 83.2	**5.05** **82.7**	**4.91** **82.2**	**4.80** **81.8**	....	....	....	....	....	....	....	
15/16	5.85 84.0	5.52 83.0	5.58 83.2	**5.50** **83.0**	**5.34** **82.4**	**5.22** **82.0**	....	....	....	....	....	....	....	
1	6.24 84.0	5.89 83.0	5.95 83.2	**5.96** **83.2**	**5.78** **82.7**	**5.63** **82.3**	**5.52** **81.9**	....	....	....	....	....	....	
1 1/16	6.63 84.0	6.26 83.0	6.32 83.2	6.36 83.3	**6.22** **83.0**	**6.06** **82.5**	**5.93** **82.1**	....	....	....	....	....	....	
1 1/8	7.02 84.0	6.62 83.0	6.69 83.2	6.74 83.3	**6.68** **83.2**	**6.50** **82.7**	**6.35** **82.3**	**6.24** **82.0**	....	....	....	....	....	
1 3/16	7.41 84.0	6.99 83.0	7.06 83.2	7.11 83.3	**7.15** **83.4**	**6.94** **82.9**	**6.78** **82.5**	**6.65** **82.1**	**6.54** **81.9**	....	....	....	....	
1 1/4	7.80 84.0	7.36 83.0	7.43 83.2	7.48 83.3	7.52 83.4	7.31 82.9	**7.22** **82.7**	**7.07** **82.3**	**6.95** **82.0**	....	....	....	....	
1 5/16	7.19 84.0	7.73 83.0	7.80 83.2	7.86 83.3	7.90 83.4	7.68 82.9	**7.67** **82.9**	**7.50** **82.5**	**7.37** **82.2**	**7.26** **81.9**	....	....	....	
1 3/8	8.58 84.0	8.09 83.0	8.18 83.2	8.23 83.3	8.27 83.4	8.04 82.9	8.09 83.0	**7.94** **82.7**	**7.79** **82.4**	**7.67** **82.1**	**7.56** **81.8**	....	....	

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by crushing on the inner row of rivets and single shearing on the outer. With a thickness of inside cover plate greater than that specified in the table, the manner of failure and hence the pitch and efficiency will not be changed.

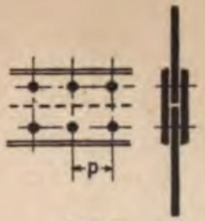
1 2 3 4 5 6 7 8 9 0

The outside cover plate thickness so long as it is substantially more than half the main plate thickness does not enter into the pitch calculation and hence it may be increased to the full main plate thickness, or even more if desired, without changing the pitch or efficiency.

TABLE X.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Single-riveted Butt-joint. — Joint D.



Joint D.

The Theoretical Pitch is not Affected so Long as Each Cover Plate is at Least Half as Thick as the Main Plate. The Inside Cover must Have an Additional Allowance for Corrosion Above This Thickness.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	Main plate thickness. Ins.
	1/4	1/4	5/16	3/8	7/16	7/16	1/2	9/16	9/16	5/8	5/8	11/16	3/4	Inside cover plate thickness. Ins.
5/8	1.72 63.6	1.72 63.6	1.72 6.36	1.72 6.36	*1.63* *61.6*	*1.52* *58.8*	*1.43* *56.2*	*1.36* *53.9*	....	....	....	....	....	
11/16	1.89 63.6	1.89 63.6	1.89 63.6	1.89 63.6	1.89 63.6	*1.77* *61.1*	*1.66* *58.6*	*1.57* *56.2*	*1.50* *54.1*	....	....	....	....	
3/4	2.06 63.6	2.06 63.6	2.06 63.6	2.06 63.6	2.06 63.6	*2.03* *63.1*	*1.91* *60.7*	*1.80* *58.4*	*1.71* *56.2*	*1.64* *54.3*	....	....	....	
13/16	2.23 63.6	2.23 63.6	2.23 63.6	2.23 63.6	2.23 63.6	2.23 63.6	*2.17* *62.5*	*2.05* *60.3*	*1.94* *58.2*	*1.86* *56.2*	*1.78* *54.4*	....	....	
7/8	2.40 63.6	2.40 63.6	2.40 63.6	2.40 63.6	2.40 63.6	2.40 63.6	2.40 63.6	*2.31* *62.1*	*2.19* *60.0*	*2.09* *58.0*	*2.00* *56.2*	*1.92* *54.6*	*1.86* *52.9*	
15/16	2.57 63.6	2.57 63.6	2.57 63.6	2.57 63.6	2.57 63.6	2.57 63.6	2.57 63.6	2.57 63.6	*2.44* *61.7*	*2.33* *59.7*	*2.23* *57.9*	*2.14* *56.2*	*2.07* *54.6*	
1	2.75 63.6	2.75 63.6	2.75 63.6	2.75 63.6	2.75 63.6	2.75 63.6	2.75 63.6	2.75 63.6	*2.71* *63.1*	*2.58* *61.3*	*2.47* *59.5*	*2.37* *57.8*	*2.28* *56.2*	
1 1/16	2.92 63.6	2.92 63.6	2.92 63.6	2.92 63.6	2.92 63.6	2.92 63.6	2.92 63.6	2.92 63.6	2.92 63.6	*2.85* *62.7*	*2.72* *61.0*	*2.61* *59.3*	*2.51* *57.7*	
1 1/8	3.09 63.6	3.09 63.6	3.09 63.6	3.09 63.6	3.09 63.6	3.09 63.6	3.09 63.6	3.09 63.6	3.09 63.6	3.09 63.6	*2.98* *62.3*	*2.86* *60.6*	*2.75* *59.1*	
1 3/16	3.26 63.6	3.26 63.6	3.26 63.6	3.26 63.6	3.26 63.6	3.26 63.6	3.26 63.6	3.26 63.6	3.26 63.6	3.26 63.6	*3.26* *63.5*	*3.12* *62.0*	*3.00* *60.4*	
1 1/4	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	3.43 63.6	*3.39* *63.1*	*3.26* *61.6*	
1 5/16	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	3.60 63.6	*3.53* *62.8*	
1 3/8	3.78	3.78	3.78	3.78	3.78	3.78	3.78	3.78	3.78	3.78	3.78	3.78	3.78	

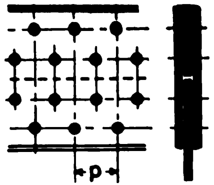
Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by double shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0



Joint E.

MAXIMUM VALUE OF PITCH AND [illegible]

Double-riveted Butt-joint with Cover Plates of Equal Width.—Joint E.

The Theoretical Pitch is not Affected so Long as Each Cover Plate is at Least Half as Thick as the Main Plate. The Inside Cover must Have an Additional Allowance for Corrosion Above This Thickness.

Steel Plate, $f_t = 55,000$ lbs. per sq. in. Steel Rivets, $f_s = 45,000$ lbs. per sq. in. $f_c = 96,000$ lbs. per sq. in.

Driven rivet diam. Ins.	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	Main plate thickness. Ins.
	1/4	1/4	5/16	3/8	7/16	7/16	1/2	9/16	9/16	5/8	5/8	11/16	3/4	Inside cover plate thickness. Ins.
5/8	2.81 77.7	2.81 77.7	2.81 77.7	2.81 77.7	*2.63* *76.3*	*2.41* *74.1*	*2.23* *72.0*	*2.08* *70.0*	*1.96* *68.2*	*1.86* *66.4*	*1.77* *64.7*	....	....	
11/16	3.09 77.7	3.09 77.7	3.09 77.7	3.09 77.7	3.09 77.7	*2.85* *75.9*	*2.63* *73.9*	*2.45* *72.0*	*2.31* *70.2*	*2.18* *68.5*	*2.08* *66.9*	*1.98* *65.3*	*1.90* *63.8*	
3/4	3.37 77.7	3.37 77.7	3.37 77.7	3.37 77.7	3.37 77.7	*3.32* *77.4*	*3.06* *75.5*	*2.85* *73.7*	*2.68* *72.0*	*2.53* *70.4*	*2.40* *68.8*	*2.29* *67.3*	*2.20* *65.9*	
13/16	3.65 77.7	3.65 77.7	3.65 77.7	3.65 77.7	3.65 77.7	3.65 77.7	*3.53* *77.0*	*3.28* *75.2*	*3.07* *73.6*	*2.90* *72.0*	*2.75* *70.5*	*2.62* *69.0*	*2.51* *67.6*	
7/8	3.93 77.7	3.93 77.7	3.93 77.7	3.93 77.7	3.93 77.7	3.93 77.7	3.93 77.7	*3.74* *76.6*	*3.50* *75.0*	*3.30* *73.5*	*3.12* *72.0*	*2.97* *70.6*	*2.84* *69.2*	
15/16	4.21 77.7	4.21 77.7	4.21 77.7	4.21 77.7	4.21 77.7	4.21 77.7	4.21 77.7	4.21 77.7	*3.95* *76.3*	*3.72* *74.8*	*3.52* *73.4*	*3.35* *72.0*	*3.20* *70.7*	
1	4.49 77.7	4.49 77.7	4.49 77.7	4.49 77.7	4.49 77.7	4.49 77.7	4.49 77.7	4.49 77.7	*4.43* *77.4*	*4.16* *76.0*	*3.94* *74.6*	*3.74* *73.3*	*3.57* *72.0*	
1 1/16	4.77 77.7	4.77 77.7	4.77 77.7	4.77 77.7	4.77 77.7	4.77 77.7	4.77 77.7	4.77 77.7	4.77 77.7	*4.63* *77.1*	*4.38* *75.7*	*4.16* *74.5*	*3.96* *73.2*	
1 1/8	5.05 77.7	5.05 77.7	5.05 77.7	5.05 77.7	5.05 77.7	5.05 77.7	5.05 77.7	5.05 77.7	5.05 77.7	5.05 77.7	*4.84* *76.8*	*4.59* *75.5*	*4.38* *74.3*	
1 3/16	5.34 77.7	5.34 77.7	5.34 77.7	5.34 77.7	5.34 77.7	5.34 77.7	5.34 77.7	5.34 77.7	5.34 77.7	5.34 77.7	*5.33* *77.7*	*5.05* *76.5*	*4.81* *75.5*	
1 1/4	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	5.61 77.7	*5.53* *77.4*	*5.26* *76.2*	
1 5/16	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	5.89 77.7	*5.74* *77.1*	
1 3/8	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	6.18 77.7	

Remarks.

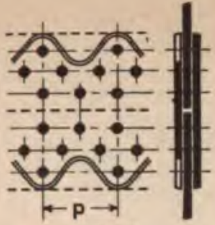
The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by double shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0

TABLE XII.



Joint F.

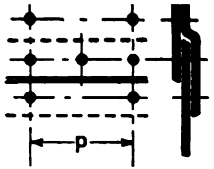
MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Triple-riveted Butt-joint with Scalloped Outside Cover Plate. — Joint F.

The Theoretical Pitch is not Affected so Long as Each Cover Plate is at Least Half as Thick as the Main Plate. The Inside Cover must Have an Additional Allowance for Corrosion Above This Thickness.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	6.08 89.7	6.08 89.7	6.08 89.7	6.08 89.7	*5.64* *88.9*	*5.09* *87.7*	*4.64* *86.5*	*4.28* *85.4*	*3.97* *84.3*	*3.71* *83.2*	*3.49* *82.1*	....	....	Remarks. The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question: 1 2 3 4 5 6 7 8 9 0 The following style of figures indicates failure by double shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable: *1 2 3 4 5 6 7 8 9 0*
$\frac{11}{16}$	6.69 89.7	6.69 89.7	6.69 89.7	6.69 89.7	6.69 89.7	*6.09* *88.7*	*5.55* *87.6*	*5.10* *86.5*	*4.74* *85.5*	*4.42* *84.5*	*4.16* *83.5*	*3.93* *82.5*	....	
$\frac{3}{4}$	7.30 89.7	7.30 89.7	7.30 89.7	7.30 89.7	7.30 89.7	*7.18* *89.6*	*6.53* *88.5*	*6.01* *87.5*	*5.57* *86.5*	*5.20* *85.6*	*4.88* *84.7*	*4.61* *83.7*	*4.36* *82.8*	
$\frac{13}{16}$	7.90 89.7	7.90 89.7	7.90 89.7	7.90 89.7	7.90 89.7	7.90 89.7	*7.60* *89.3*	*6.98* *88.4*	*6.47* *87.4*	*6.03* *86.5*	*5.66* *85.6*	*5.34* *84.8*	*5.05* *83.9*	
$\frac{7}{8}$	8.51 89.7	8.51 89.7	8.51 89.7	8.51 89.7	8.51 89.7	8.51 89.7	8.51 89.7	*8.03* *89.1*	*7.43* *88.2*	*6.93* *87.4*	*6.50* *86.5*	*6.12* *85.7*	*5.79* *84.9*	
$\frac{15}{16}$	9.11 89.7	9.11 89.7	9.11 89.7	9.11 89.7	9.11 89.7	9.11 89.7	9.11 89.7	9.11 89.7	*8.47* *88.9*	*7.89* *88.1*	*7.39* *87.3*	*6.96* *86.5*	*6.58* *85.8*	
1	9.73 89.7	9.73 89.7	9.73 89.7	9.73 89.7	9.73 89.7	9.73 89.7	9.73 89.7	9.73 89.7	*9.57* *89.5*	*8.91* *88.8*	*8.34* *88.0*	*7.86* *87.3*	*7.43* *86.5*	
$1\frac{1}{16}$	10.34 89.7	10.34 89.7	10.34 89.7	10.34 89.7	10.34 89.7	10.34 89.7	10.34 89.7	10.34 89.7	10.34 89.7	*9.99* *89.4*	*9.35* *88.7*	*8.80* *87.9*	*8.31* *87.2*	
$1\frac{1}{8}$	10.94 89.7	10.94 89.7	10.94 89.7	10.94 89.7	10.94 89.7	10.94 89.7	10.94 89.7	10.94 89.7	10.94 89.7	10.94 89.7	*10.42* *89.2*	*9.80* *88.5*	*9.26* *87.9*	
$1\frac{3}{16}$	11.56 89.7	11.56 89.7	11.56 89.7	11.56 89.7	11.56 89.7	11.56 89.7	11.56 89.7	11.56 89.7	11.56 89.7	11.56 89.7	*11.54* *89.7*	*10.86* *89.0*	*10.25* *88.4*	
$1\frac{1}{4}$	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	12.16 89.7	*11.96* *89.6*	*11.28* *89.0*	
$1\frac{5}{16}$	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.77 89.7	12.38 *89.4*	
$1\frac{3}{8}$	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	13.38 89.7	



Joint G.

TABLE XIII.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Single-riveted Welt-joint. — Joint G.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	*3.64* *82.8*	*3.13* *80.0*	....	....	....	....	....	....	....	....	....	....	....	
$\frac{11}{16}$	4.29 84.0	**3.59** **80.8**	*3.12* *77.9*	....	....	....	....	....	....	....	....	....	....	
$\frac{3}{4}$	4.68 84.0	**4.11** **81.8**	*3.64* *79.4*	*3.23* *76.8*	....	....	....	....	....	....	....	....	....	
$\frac{13}{16}$	5.07 84.0	**4.66** **82.5**	*4.21* *80.7*	*3.72* *78.2*	....	....	....	....	....	....	....	....	....	
$\frac{7}{8}$	5.46 84.0	5.15 83.0	**4.77** **81.7**	*4.25* *79.4*	*3.83* *77.1*	....	....	....	....	....	....	....	....	
$\frac{15}{16}$	5.84 84.0	5.52 83.0	**5.31** **82.4**	*4.81* *80.5*	*4.32* *78.3*	....	....	....	....	....	....	....	....	
1	6.24 84.0	5.89 83.0	**5.88** **83.0**	*5.41* *81.5*	*4.86* *79.4*	*4.43* *77.4*	....	....	....	....	....	....	....	
$1\frac{1}{16}$	6.63 84.0	6.26 83.0	6.32 83.2	**5.97** **82.2**	*5.42* *80.4*	*4.93* *78.5*	*4.55* *76.6*	....	....	....	....	....	....	
$1\frac{1}{8}$	7.01 84.0	6.62 83.0	6.69 83.2	**6.53** **82.8**	*6.01* *81.3*	*5.46* *79.4*	*5.03* *77.6*	....	....	....	....	....	....	
$1\frac{3}{16}$	7.41 84.0	6.99 83.0	7.06 83.2	**7.11** **83.8**	*6.62* *82.1*	*6.02* *80.3*	*5.54* *78.6*	*5.14* *76.9*	....	....	....	....	....	
$1\frac{1}{4}$	7.80 84.0	7.36 83.0	7.43 83.2	7.48 83.3	**7.17** **82.6**	*6.60* *81.8*	*6.07* *79.4*	*5.63* *77.8*	*5.27* *76.3*	....	....	....	....	
$1\frac{5}{16}$	8.19 84.0	7.73 83.0	7.80 83.2	7.86 83.3	**7.75** **83.1**	*7.22* *81.8*	*6.63* *80.2*	*6.14* *78.6*	*5.74* *77.1*	....	....	....	....	
$1\frac{3}{8}$	8.58 84.0	8.10 83.0	8.18 83.2	8.23 83.3	8.28 83.4	**7.56** **81.8**	**7.18** **80.9**	*6.68* *79.4*	*6.24* *77.9*	*5.86* *76.5*	....	....	....	

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

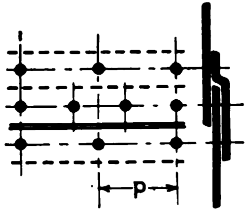
The following style of figures indicates failure by single shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by single shearing on the inner row of rivets and crushing on the outer.

1 2 3 4 5 6 7 8 9 0

For the effect produced by varying the inside cover plate thickness, see page 156.



Joint H.

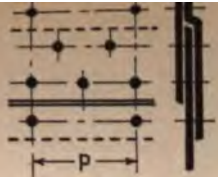
TABLE XIV.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Single-riveted Welt-joint. — Joint H.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	*3.14* *80.1*	*2.63* *76.2*	*2.30* *72.8*	....	....	....	....	....	....	....	....	....	....	**Remarks.**
$\frac{11}{16}$	3.69 81.4	**3.10** **77.8**	*2.71* *74.7*	*2.42* *71.6*	....	....	....	....	....	....	....	....	....	**The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:**
$\frac{3}{4}$	4.02 81.4	**3.53** **78.8**	*3.16* *76.3*	*2.82* *73.4*	*2.56* *70.7*	....	....	....	....	....	....	....	....	
$1\frac{3}{16}$	4.35 81.4	**3.98** **79.6**	*3.62* *77.6*	*3.24* *74.9*	*2.93* *72.3*	*2.70* *69.9*	....	....	....	....	....	....	....	1 2 3 4 5 6 7 8 9 0
$\frac{7}{8}$	4.69 81.4	4.39 80.1	**4.12** **78.7**	*3.69* *76.3*	*3.33* *73.8*	*3.06* *71.4*	*2.84* *69.3*	....	....	....	....	....	....	**The following style of figures indicates failure by single shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable:**
$1\frac{5}{16}$	5.03 81.4	4.70 80.1	**4.56** **79.4**	*4.16* *77.5*	*3.76* *75.1*	*3.45* *72.8*	*3.20* *70.7*	*2.99* *68.6*	....	....	....	....	....	
1	5.37 81.4	5.02 80.1	**5.03** **80.1**	*4.67* *78.6*	*4.21* *76.3*	*3.86* *74.0*	*3.57* *72.0*	*3.34* *70.0*	....	....	....	....	....	*1 2 3 4 5 6 7 8 9 0*
$1\frac{1}{16}$	5.70 81.4	5.33 80.1	5.39 80.3	**5.14** **79.3**	*4.69* *77.4*	*4.29* *75.2*	*3.97* *73.2*	*3.70* *71.3*	*3.48* *69.5*	....	....	....	....	**The following style of figures indicates failure by single shearing on the inner row of rivets and crushing on the outer:**
$1\frac{1}{8}$	6.03 81.4	5.64 80.1	5.71 80.3	**5.60** **79.9**	*5.19* *78.3*	*4.74* *76.3*	*4.38* *74.3*	*4.08* *72.4*	*3.84* *70.7*	*3.63* *69.0*	....	....	....	
$1\frac{3}{16}$	6.37 81.4	5.96 80.1	6.03 80.3	**6.07** **80.4**	*5.72* *79.2*	*5.22* *77.3*	*4.81* *75.3*	*4.48* *73.5*	*4.21* *71.8*	*3.98* *70.1*	*3.76* *68.6*	....	....	**1 2 3 4 5 6 7 8 9 0**
$1\frac{1}{4}$	6.71 81.4	6.27 80.1	6.34 80.3	6.39 80.5	**6.17** **79.8**	*5.71* *78.1*	*5.27* *76.3*	*4.90* *74.5*	*4.60* *72.8*	*4.34* *71.2*	*4.12* *69.7*	*3.93* *68.2*	....	**For the effect produced by varying the inside cover plate thickness, see page 156.**
$1\frac{5}{16}$	7.04 81.4	6.58 80.1	6.66 80.3	6.71 80.5	**6.64** **80.2**	*6.23* *78.9*	*5.74* *77.1*	*5.34* *75.4*	*5.00* *73.7*	*4.72* *72.2*	*4.48* *70.7*	*4.27* *69.2*	....	
$1\frac{3}{8}$	7.38 81.4	6.90 80.1	6.98 80.3	7.03 80.5	7.08 80.6	**6.48** **78.8**	**6.21** **77.8**	*5.79* *76.8*	*5.43* *74.7*	*5.11* *73.1*	*4.85* *71.6*	*4.62* *70.6*	*4.41* *68.8*	



Joint I.

TABLE XV.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Double-riveted Welt-joint. — Joint I.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	Main plate thickness. Ins.
	1/4	1/4	5/16	3/8	7/16	7/16	1/2	9/16	9/16	5/8	5/8	11/16	3/4	Inside cover plate thickness. Ins.
5/8	*5.65* *88.9*	*4.64* *86.5*	*3.97* *84.3*	*3.49* *82.1*	*3.13* *80.1*	*2.86* *78.1*	....	....	....	....	....	....	....	
11/16	6.69 89.7	**5.53** **87.6**	*4.74* *85.5*	*4.16* *83.5*	*3.72* *81.6*	*3.39* *79.7*	*3.12* *77.9*	....	....	....	....	....	....	
3/4	7.30 89.7	**6.42** **88.3**	*5.57* *86.5*	*4.88* *84.7*	*4.36* *82.8*	*3.96* *81.1*	*3.64* *79.4*	*3.38* *77.8*	....	....	....	....	....	
13/16	7.90 89.7	**7.37** **89.0**	*6.47* *87.4*	*5.66* *85.6*	*5.05* *83.9*	*4.58* *82.2*	*4.21* *80.7*	*3.90* *79.2*	*3.64* *77.7*	....	....	....	....	
7/8	8.51 89.7	8.21 89.3	**7.40** **88.2**	*6.50* *86.5*	*5.79* *84.9*	*5.25* *83.3*	*4.81* *81.8*	*4.45* *80.4*	*4.15* *78.9*	*3.90* *77.6*	....	....	....	
15/16	9.11 89.7	8.79 89.3	**8.32** **88.7**	*7.39* *87.3*	*6.58* *85.8*	*5.96* *84.3*	*5.46* *82.8*	*5.04* *81.4*	*4.70* *80.1*	*4.41* *78.8*	*4.16* *77.5*	....	....	
1	9.73 89.7	9.38 89.3	**9.31** **89.3**	*8.35* *88.0*	*7.43* *86.5*	*6.71* *85.1*	*6.14* *83.7*	*5.67* *82.4*	*5.28* *81.1*	*4.95* *79.8*	*4.67* *78.6*	*4.43* *77.4*	....	
1 1/16	10.34 89.7	9.97 89.3	10.03 89.4	**9.28** **88.6**	*8.32* *87.2*	*7.51* *85.8*	*6.88* *84.5*	*6.33* *83.2*	*5.90* *82.0*	*5.53* *80.8*	*5.21* *81.4*	*4.93* *78.5*	*4.69* *77.3*	
1 1/8	10.94 89.7	10.55 89.3	10.61 89.4	**10.24** **89.0**	*9.26* *87.9*	*8.35* *86.5*	*7.63* *85.3*	*7.04* *84.0*	*6.55* *82.8*	*6.13* *81.6*	*5.77* *80.5*	*5.46* *79.4*	*5.19* *78.3*	
1 3/16	11.56 89.7	11.14 89.3	11.21 89.4	**11.25** **89.5**	*10.25* *88.4*	*9.24* *87.2*	*8.44* *85.9*	*7.78* *84.7*	*7.23* *83.6*	*6.76* *82.5*	*6.36* *81.3*	*6.02* *80.3*	*5.72* *79.3*	
1 1/4	12.16 89.7	11.72 89.3	11.80 89.4	11.85 89.5	**11.19** **88.8**	*10.17* *87.9*	*9.28* *86.5*	*8.55* *85.4*	*7.95* *84.3*	*7.43* *83.2*	*6.99* *82.1*	*6.61* *81.1*	*6.27* *80.0*	
1 5/16	12.77 89.7	12.31 89.3	12.39 89.4	12.44 89.5	**12.17** **89.3**	*11.15* *88.3*	*10.17* *87.1*	*9.36* *86.0*	*8.69* *84.9*	*8.12* *83.8*	*7.64* *82.8*	*7.22* *81.8*	*6.85* *80.8*	
1 3/8	13.38 89.7	12.90 89.3	12.98 89.4	13.03 89.5	13.08 89.5	**11.88** **88.5**	**11.07** **87.6**	*10.21* *86.5*	*9.48* *85.5*	*8.85* *84.5*	*8.32* *83.4*	*7.86* *82.5*	*7.45* *81.6*	

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

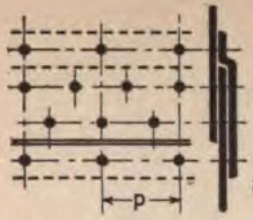
The following style of figures indicates failure by single shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by single shearing on the inner row of rivets and crushing on the outer:

1 2 3 4 5 6 7 8 9 0

For the effect produced by varying the inside cover plate thickness, see page 156.



Joint J.

TABLE XVI.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Double-riveted Welt-joint. — Joint J.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	Main plate thickness. Ins.
	1/4	1/4	5/16	3/8	7/16	7/16	1/2	9/16	9/16	5/8	5/8	11/16	3/4	Inside cover plate thickness. Ins.
5/8	*4.64* *86.5*	*3.84* *83.7*	*3.30* *81.1*	*2.92* *78.6*	*2.69* *74.5*	....	....	....	....	....	....	....	....	
11/16	5.49 87.5	**4.56** **84.9**	*3.93* *82.5*	*3.46* *80.1*	*3.12* *77.9*	*2.85* *75.9*	....	....	....	....	....	....	....	
3/4	5.99 87.5	**5.27** **85.8**	*4.61* *83.7*	*4.05* *81.5*	*3.64* *79.4*	*3.32* *77.4*	....	....	....	....	....	....	....	
13/16	6.48 87.5	**6.02** **86.5**	*5.33* *84.8*	*4.69* *82.7*	*4.21* *80.7*	*3.83* *78.8*	*3.53* *77.0*	....	....	....	....	....	....	
7/8	6.98 87.5	6.68 86.9	**6.08** **85.6**	*5.37* *83.7*	*4.81* *81.8*	*4.37* *80.0*	*4.02* *78.3*	*3.74* *76.6*	....	....	....	....	....	
15/16	7.48 87.5	7.15 86.9	**6.82** **86.3**	*6.10* *84.6*	*5.45* *82.8*	*4.95* *81.0*	*4.55* *79.4*	*4.22* *77.8*	*3.95* *76.3*	....	....	....	....	
1	7.98 87.5	7.63 86.9	**7.60** **86.9**	*6.88* *85.4*	*6.14* *83.7*	*5.57* *82.1*	*5.11* *80.4*	*4.74* *78.9*	*4.43* *77.4*	*4.16* *75.7*	....	....	....	
1 1/16	8.48 87.5	8.11 86.9	8.17 87.0	**7.63** **86.1**	*6.87* *84.5*	*6.22* *82.9*	*5.71* *81.4*	*5.28* *79.9*	*4.93* *78.5*	*4.63* *77.1*	*4.38* *75.7*	....	....	
1 1/8	8.98 87.5	8.58 86.9	8.65 87.0	**8.39** **86.6**	*7.63* *85.3*	*6.91* *83.7*	*6.33* *82.2*	*5.85* *80.8*	*5.46* *79.4*	*5.13* *78.1*	*4.84* *76.8*	*4.60* *75.5*	....	
1 3/16	9.48 87.5	9.07 86.9	9.14 87.0	**9.18** **87.0**	*8.44* *85.9*	*7.63* *84.4*	*6.99* *83.0*	*6.46* *81.6*	*6.02* *80.3*	*5.65* *79.0*	*5.33* *77.7*	*5.05* *76.5*	*4.81* *75.3*	
1 1/4	9.98 87.5	9.54 86.9	9.62 87.0	9.67 87.1	**9.18** **86.4**	*8.39* *85.1*	*7.67* *83.7*	*7.09* *82.4*	*6.61* *81.1*	*6.19* *79.8*	*5.84* *78.6*	*5.53* *77.4*	*5.26* *76.2*	
1 5/16	10.48 87.5	10.02 86.9	10.10 87.0	10.15 87.1	**9.96** **86.8**	*9.18* *85.7*	*8.40* *84.3*	*7.75* *83.1*	*7.22* *81.8*	*6.76* *80.6*	*6.37* *79.4*	*6.04* *78.3*	*5.74* *77.1*	
	10.98	10.50	10.58	10.63	10.68	**9.72**	**9.13**	*8.44*	*7.86*	*7.36*	*6.93*	*6.56*	*6.24*	

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

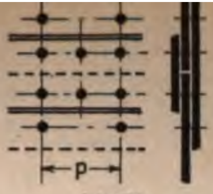
The following style of figures indicates failure by single shearing throughout the joint and shows pitches and efficiencies somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by single shearing on the inner rows of rivets and crushing on the outer:

1 2 3 4 5 6 7 8 9 0

For the effect produced by varying the inside cover plate thickness, see page 156.



Joint K.

TABLE XVII.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Double-riveted Butt-joint. — Joint K.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	**3.81** **83.6**	**3.61** **82.7**	**3.48** **82.0**	....	....	....	....	....	....	....	....	....	....	
$\frac{11}{16}$	4.29 84.0	4.05 83.0	**3.90** **82.4**	**3.78** **81.8**	....	....	....	....	....	....	....	....	....	
$\frac{3}{4}$	4.68 84.0	4.42 83.0	**4.33** **82.7**	**4.19** **82.1**	....	....	....	....	....	....	....	....	....	
$\frac{13}{16}$	5.07 84.0	4.78 83.0	**4.78** **83.0**	**4.62** **82.4**	**4.50** **81.9**	....	....	....	....	....	....	....	....	
$\frac{7}{8}$	5.46 84.0	5.15 83.0	5.20 83.2	**5.05** **82.7**	**4.91** **82.2**	**4.80** **81.8**	....	....	....	....	....	....	....	
$\frac{15}{16}$	5.85 84.0	5.52 83.0	5.58 83.2	**5.50** **83.0**	**5.34** **82.4**	**5.22** **82.0**	....	....	....	....	....	....	....	
1	6.24 84.0	5.89 83.0	5.95 83.2	**5.96** **83.2**	**5.78** **82.7**	**5.63** **82.3**	**5.52** **81.9**	....	....	....	....	....	....	
$1\frac{1}{16}$	6.63 84.0	6.26 83.0	6.32 83.2	6.36 83.3	**6.22** **83.0**	**6.06** **82.5**	**5.93** **82.1**	....	....	....	....	....	....	
$1\frac{1}{8}$	7.02 84.0	6.62 83.0	6.69 83.2	6.74 83.3	**6.68** **83.2**	**6.50** **82.7**	**6.35** **82.3**	**6.24** **82.0**	....	....	....	....	....	
$1\frac{3}{16}$	7.41 84.0	6.99 83.0	7.06 83.2	7.11 83.3	**7.15** **83.4**	**6.94** **82.9**	**6.78** **82.5**	**6.65** **82.1**	**6.54** **81.9**	....	....	....	....	
$1\frac{1}{4}$	7.80 84.0	7.36 83.0	7.43 83.2	7.48 83.3	7.52 83.4	7.31 82.9	**7.22** **82.7**	**7.07** **82.3**	**6.95** **82.0**	....	....	....	....	
$1\frac{5}{16}$	7.19 84.0	7.73 83.0	7.80 83.2	7.86 83.3	7.90 83.4	7.68 82.9	**7.67** **82.9**	**7.50** **82.5**	**7.37** **82.2**	**7.26** **81.9**	....	....	....	
$1\frac{3}{8}$	8.58 84.0	8.09 83.0	8.18 83.2	8.23 83.3	8.27 83.4	8.04 82.9	8.09 83.0	**7.94** **82.7**	**7.79** **82.4**	**7.67** **82.1**	**7.56** **81.8**	....	....	

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by crushing on the inner row of rivets and single shearing on the outer. With a thickness of inside cover plate greater than that specified in the table, the manner of failure and hence the pitch and efficiency will not be changed.

1 2 3 4 5 6 7 8 9 0

The outside cover plate thickness so long as it is substantially more than half the main plate thickness does not enter into the pitch calculation and hence it may be increased to the full main plate thickness, or even more if desired, without changing the pitch or efficiency.

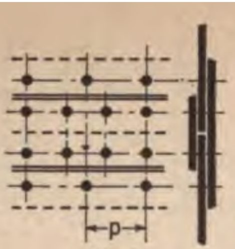

Joint L.

TABLE XVIII.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Double-riveted Butt-joint. — Joint L.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in. $f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	**3.27** **80.9**	**3.07** **79.6**	**2.93** **78.7**	**2.84** **78.0**	*2.63* *76.3*	*2.41* *74.1*	*2.23* *72.0*	*2.09* *70.0*	....	....	....	....	....	
$\frac{11}{16}$	3.69 81.4	3.45 80.1	**3.30** **79.1**	**3.18** **78.4**	**3.09** **77.8**	*2.85* *75.8*	*2.63* *73.8*	*2.46* *72.0*	*2.31* *70.2*	....	....	....	....	
$\frac{3}{4}$	4.02 81.4	3.76 80.1	**3.68** **79.6**	**3.54** **78.8**	**3.44** **78.2**	*3.32* *77.4*	*3.06* *75.5*	*2.85* *73.7*	*2.68* *72.0*	*2.53* *70.4*	....	....	....	
$\frac{13}{16}$	4.36 81.4	4.07 80.1	**4.07** **80.0**	**3.91** **79.2**	**3.79** **78.5**	**3.69** **78.0**	*3.53* *77.0*	*3.28* *75.2*	*3.08* *73.6*	*2.90* *72.0*	*2.75* *70.5*	*2.62* *69.0*	....	
$\frac{7}{8}$	4.69 81.4	4.39 80.1	4.44 80.3	**4.29** **79.6**	**4.15** **78.9**	**4.04** **78.3**	**3.95** **77.9**	*3.74* *76.6*	*3.50* *75.0*	*3.30* *73.5*	*3.12* *72.0*	*2.97* *70.6*	*2.84* *69.2*	
$\frac{15}{16}$	5.03 81.4	4.70 80.1	4.76 80.3	**4.68** **80.0**	**4.52** **79.3**	**4.40** **78.7**	**4.30** **78.2**	**4.21** **77.7**	*3.95* *76.3*	*3.72* *74.8*	*3.52* *73.4*	*3.35* *72.0*	*3.20* *70.7*	
1	5.36 81.4	5.01 80.1	5.07 80.3	**5.09** **80.3**	**4.90** **79.6**	**4.76** **79.0**	**4.65** **78.5**	**4.55** **78.0**	*4.43* *77.4*	*4.16* *76.0*	*3.94* *74.6*	*3.74* *73.3*	*3.57* *72.0*	
$1\frac{1}{16}$	5.70 81.4	5.33 80.1	5.39 80.3	5.43 80.5	**5.30** **79.9**	**5.13** **79.3**	**5.00** **78.8**	**4.90** **78.3**	**4.81** **77.9**	*4.63* *77.1*	*4.38* *75.7*	*4.16* *74.4*	*3.96* *73.2*	
$1\frac{1}{8}$	6.04 81.4	5.64 80.1	5.71 80.3	5.75 80.5	**5.70** **80.2**	**5.52** **79.6**	**5.37** **79.1**	**5.26** **78.6**	**5.16** **78.2**	**5.07** **77.8**	*4.84* *76.8*	*4.60* *75.5*	*4.38* *74.3*	
$1\frac{3}{16}$	6.37 81.4	5.95 80.1	6.02 80.3	6.07 80.5	**6.11** **80.6**	**5.91** **79.9**	**5.75** **79.3**	**5.62** **78.8**	**5.51** **78.4**	**5.41** **78.0**	*5.33* *77.7*	*5.05* *76.5*	*4.81* *75.3*	
$1\frac{1}{4}$	6.71 81.4	6.27 80.1	6.34 80.3	6.39 80.5	6.43 80.6	6.22 79.9	**6.13** **79.6**	**5.98** **79.1**	**5.86** **78.7**	**5.76** **78.3**	**5.67** **78.0**	*5.53* *77.4*	*5.27* *76.3*	
$1\frac{5}{16}$	7.04 81.4	6.58 80.1	6.66 80.3	6.71 80.5	6.75 80.6	6.53 79.9	**6.52** **79.9**	**6.36** **79.3**	**6.22** **78.9**	**6.11** **78.5**	**6.01** **78.2**	**5.93** **77.9**	*5.74* *77.1*	
$1\frac{3}{8}$	7.38 81.4	6.89 80.1	6.98 80.3	7.03 80.5	7.08 80.6	6.84 79.9	6.89 80.1	**6.74** **79.6**	**6.60** **79.3**	**6.47** **78.7**	**6.36** **78.4**	**6.27** **78.1**	**6.19** **77.8**	

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

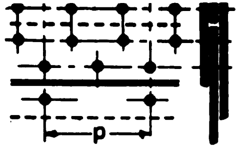
The following style of figures indicates failure by double shear on the inside row and single shear on the outside row. The pitches and efficiencies are somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by crushing on the inner row of rivets and single shearing on the outer. With a thickness of inside cover plate greater than that specified in the table the manner of failure and hence the pitch and efficiency will not be changed.

1 2 3 4 5 6 7 8 9 0

The outside cover plate thickness so long as it is substantially more than half the main plate thickness does not enter into the pitch calculation and hence it may be increased to the full main plate thickness, or even more if desired, without changing the pitch or efficiency.



Joint M.

TABLE XIX.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Triple-riveted Butt-joint. — Joint M.

Steel Plate, f_t = 55,000 lbs. per sq. in. Steel Rivets, f_s = 45,000 lbs. per sq. in.
f_c = 96,000 lbs. per sq. in.

Driven rivet diam. Ins.	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	Main plate thickness. Ins.
	1/4	1/4	5/16	3/8	7/16	7/16	1/2	9/16	9/16	5/8	5/8	11/16	3/4	Inside cover plate thickness. Ins.
5/8	**5.99** **89.6**	**5.79** **89.2**	**5.66** **89.0**	....	....	....	....	....	....	....	....	....	....	
11/16	6.69 89.7	6.45 89.3	**6.30** **89.1**	**6.18** **88.9**	....	....	....	....	....	....	....	....	....	
3/4	7.30 89.7	7.03 89.3	**6.95** **89.2**	**6.81** **89.0**	....	....	....	....	....	....	....	....	....	
13/16	7.90 89.7	7.62 89.3	**7.62** **89.3**	**7.45** **89.1**	**7.33** **88.9**	....	....	....	....	....	....	....	....	
7/8	8.51 89.7	8.21 89.3	8.26 89.4	**8.11** **89.2**	**7.97** **89.0**	**7.86** **88.8**	....	....	....	....	....	....	....	
15/16	9.12 89.7	8.79 89.3	8.85 89.4	**8.77** **89.3**	**8.61** **89.1**	**8.49** **88.9**	....	....	....	....	....	....	....	
1	9.73 89.7	9.38 89.3	9.44 89.4	**9.45** **89.4**	**9.27** **89.2**	**9.12** **89.0**	**9.01** **88.9**	....	....	....	....	....	....	
1 1/16	10.33 89.7	9.96 89.3	10.03 89.4	10.07 89.5	**9.93** **89.3**	**9.77** **89.1**	**9.64** **89.0**	....	....	....	....	....	....	
1 1/8	10.95 89.7	10.55 89.3	10.62 89.4	10.66 89.5	**10.61** **89.4**	**10.43** **89.2**	**10.28** **89.1**	**10.16** **88.9**	....	....	....	....	....	
1 3/16	11.55 89.7	11.14 89.3	11.21 89.4	11.25 89.5	**11.29** **89.5**	**11.09** **89.3**	**10.93** **89.1**	**10.80** **89.0**	**10.69** **88.9**	....	....	....	....	
1 1/4	12.16 89.7	11.72 89.3	11.80 89.4	11.85 89.5	11.89 89.5	11.67 89.3	**11.59** **89.2**	**11.44** **89.1**	**11.32** **88.9**	....	....	....	....	
1 5/16	12.77 89.7	12.31 89.3	12.38 89.4	12.44 89.5	12.48 89.5	12.26 89.3	**12.25** **89.3**	**12.09** **89.1**	**11.95** **89.0**	**11.84** **88.9**	....	....	....	
1 3/8	13.38 89.7	12.90 89.3	12.98 89.4	13.03 89.5	13.08 89.5	12.85 89.3	12.90 89.3	**12.75** **89.2**	**12.60** **89.1**	**12.47** **89.0**	**12.37** **88.9**	....	....	

Remarks.

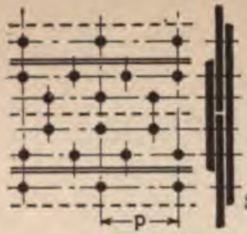
The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by crushing on the inner rows of rivets and single shearing on the outer. With a thickness of inside cover plate greater than that specified in the table, the manner of failure and hence the pitch and efficiency will not be changed:

1 2 3 4 5 6 7 8 9 0

The outside cover plate thickness so long as it is substantially more than half the main plate thickness does not enter into the pitch calculation and hence it may be increased to the full main plate thickness, or even more if desired, without changing the pitch or efficiency.



Joint N.

TABLE XX.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Triple-riveted Butt-joint. — Joint N.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	**4.90** **87.2**	**4.70** **86.7**	**4.57** **86.3**	**4.47** **86.0**	*4.14* *84.9*	*3.75* *83.4*	*3.44* *81.8*	*3.18* *80.4*	*2.97* *78.9*	*2.79* *77.6*	....	....	....	
$\frac{11}{16}$	5.49 87.5	5.25 86.9	**5.10** **86.5**	**4.98** **86.2**	**4.89** **85.9**	*4.47* *84.6*	*4.09* *83.2*	*3.78* *81.8*	*3.52* *80.5*	*3.30* *79.2*	*3.12* *77.9*	....	....	
$\frac{3}{4}$	5.99 87.5	5.73 86.9	**5.64** **86.7**	**5.50** **86.4**	**5.40** **86.1**	*5.25* *85.7*	*4.80* *84.4*	*4.43* *83.1*	*4.12* *81.8*	*3.86* *80.6*	*3.64* *79.4*	*3.45* *78.2*	....	
$\frac{13}{16}$	6.49 87.5	6.20 86.9	**6.20** **86.9**	**6.04** **86.5**	**5.91** **86.3**	**5.82** **86.0**	*5.56* *85.4*	*5.13* *84.2*	*4.77* *83.0*	*4.47* *81.8*	*4.21* *80.7*	*3.98* *79.6*	*3.78* *78.5*	
$\frac{7}{8}$	6.98 87.5	6.68 86.9	6.73 87.0	**6.58** **86.7**	**6.44** **86.4**	**6.33** **86.2**	**6.24** **86.0**	*5.89* *85.1*	*5.47* *84.0*	*5.11* *82.9*	*4.81* *81.8*	*4.55* *80.8*	*4.31* *79.7*	
$\frac{15}{16}$	7.48 87.5	7.16 86.9	7.21 87.0	**7.14** **86.9**	**6.98** **86.6**	**6.85** **86.3**	**6.75** **86.1**	**6.67** **85.9**	*6.21* *84.9*	*5.80* *83.8*	*5.46* *82.8*	*5.15* *81.8*	*4.89* *80.8*	
1	7.96 87.5	7.63 86.9	7.69 87.0	**7.71** **87.0**	**7.52** **86.7**	**7.38** **86.4**	**7.27** **86.2**	**7.17** **86.1**	*7.00* *85.7*	*6.54* *84.7*	*6.14* *83.7*	*5.80* *82.8*	*5.50* *81.8*	
$1\frac{1}{16}$	8.48 87.5	8.11 86.9	8.17 87.0	8.21 87.1	**8.08** **86.8**	**7.92** **86.6**	**7.79** **86.3**	**7.68** **86.2**	**7.59** **86.0**	*7.31* *85.5*	*6.87* *84.5*	*6.48* *83.6*	*6.14* *82.7*	
$1\frac{1}{8}$	8.98 87.5	8.59 86.9	8.65 87.0	8.70 87.1	**8.65** **87.0**	**8.46** **86.7**	**8.32** **86.5**	**8.20** **86.3**	**8.10** **86.1**	**8.02** **86.0**	*7.63* *85.3*	*7.20* *84.4*	*6.82* *83.5*	
$1\frac{3}{16}$	9.48 87.5	9.06 86.9	9.13 87.0	9.18 87.1	**9.22** **87.1**	**9.02** **86.8**	**8.86** **86.6**	**8.72** **86.4**	**8.62** **86.2**	**8.52** **86.1**	*8.44* *85.9*	*7.95* *85.1*	*7.53* *84.2*	
$1\frac{1}{4}$	9.98 87.5	9.54 86.9	9.61 87.0	9.66 87.1	9.70 87.1	9.49 86.8	**9.40** **86.7**	**9.26** **86.5**	**9.14** **86.3**	**9.03** **86.2**	**8.94** **86.0**	*8.75* *85.7*	*8.28* *84.9*	
$1\frac{5}{}$	10.48	10.02	10.09	10.15	10.19	9.97	**9.96**	**9.80**	**9.66**	**9.55**	**9.45**	**9.37**	*9.06*	

Remarks.

The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:

1 2 3 4 5 6 7 8 9 0

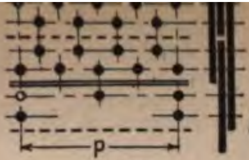
The following style of figures indicates failure by double shear on the inside rows of rivets and single shear on the outside row. The pitches and efficiencies are somewhat less than the maximum obtainable:

1 2 3 4 5 6 7 8 9 0

The following style of figures indicates failure by crushing on the inner rows of rivets and single shearing on the outer. With a thickness of inside cover plate greater than that specified in the table, the manner of failure, and hence the pitch and efficiency, will not be changed.

1 2 3 4 5 6 7 8 9 0

The outside cover plate thickness so long as it is substantially more than half the main plate thickness does not enter into the pitch calculation and hence it may be increased to the full main plate thickness, or



Joint O.

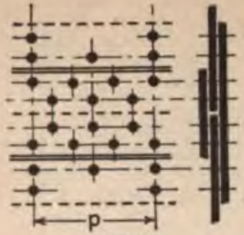
TABLE XXI.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Quadruple-riveted Butt-joint. — Joint O.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	**12.36** **95.0**	**11.76** **94.7**	**11.36** **94.6**	....	....	....	....	....	....	....	....	....	....	Remarks.
$\frac{11}{16}$	13.89 95.1	13.17 94.8	**12.72** **94.6**	**12.37** **94.5**	....	....	....	....	....	....	....	....	....	The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:
$\frac{3}{4}$	15.15 95.1	14.36 94.8	**14.11** **94.7**	**13.70** **94.5**	....	....	....	....	....	....	....	....	....	
$\frac{13}{16}$	16.42 95.1	15.56 94.8	**15.55** **94.7**	**15.06** **94.6**	**14.70** **94.5**	....	....	....	....	....	....	....	....	1 2 3 4 5 6 7 8 9 0
$\frac{7}{8}$	17.68 95.1	16.76 94.8	16.91 94.8	**16.47** **94.7**	**16.05** **94.5**	**15.72** **94.4**	....	....	....	....	....	....	....	The following style of figures indicates failure by crushing on the inner rows of rivets and single shearing on the outer. With a thickness of inside cover plate greater than that specified in the table, the manner of failure, and hence the pitch and efficiency, will not be changed.
$\frac{15}{16}$	18.94 95.1	17.95 94.8	18.10 94.8	**17.90** **94.8**	**17.42** **94.6**	**17.04** **94.5**	....	....	....	....	....	....	....	
1	20.20 95.1	19.15 94.8	19.31 94.8	**19.37** **94.8**	**18.82** **94.7**	**18.39** **94.5**	**18.04** **94.5**	....	....	....	....	....	....	
$1\frac{1}{16}$	21.47 95.1	20.36 94.8	20.52 94.8	20.67 94.9	**20.26** **94.8**	**19.78** **94.6**	**19.39** **94.5**	....	....	....	....	....	....	**1 2 3 4 5 6 7 8 9 0**
$1\frac{1}{8}$	22.73 95.1	21.55 94.8	21.73 94.8	21.88 94.9	**21.70** **94.8**	**21.16** **94.7**	**20.73** **94.6**	**20.37** **94.5**	....	....	....	....	....	
$1\frac{3}{16}$	24.00 95.1	22.76 94.8	22.95 94.8	23.12 94.9	**23.21** **94.9**	**22.61** **94.7**	**22.13** **94.6**	**21.73** **94.5**	**21.40** **94.5**	....	....	....	....	The outside cover plate thickness so long as it is substantially more than half the main plate thickness does not enter into the pitch calculation and hence it may be increased to the full main plate thickness, or even more if desired, without changing the pitch or efficiency.
$1\frac{1}{4}$	25.25 95.1	23.94 94.8	24.14 94.8	24.31 94.9	24.42 94.9	23.79 94.7	**23.53** **94.7**	**23.09** **94.6**	**22.73** **94.5**	....	....	....	....	
$1\frac{5}{16}$	26.53 95.1	25.14 94.8	25.35 94.8	25.54 94.9	25.65 94.9	24.99 94.7	**24.96** **94.7**	**24.47** **94.6**	**24.07** **94.6**	**23.73** **94.5**	....	....	....	
$1\frac{3}{8}$	27.78 95.1	26.34 94.8	26.56 94.8	26.75 94.9	26.87 94.9	26.16 94.7	26.34 94.7	**25.88** **94.7**	**25.44** **94.6**	**25.06** **94.6**	**24.74** **94.5**	....	....	



Joint P.

TABLE XXII.

MAXIMUM VALUE OF PITCH AND EFFICIENCY.

Quadruple-riveted Butt-joint. — Joint P.

Steel Plate, $f_t = 55{,}000$ lbs. per sq. in. Steel Rivets, $f_s = 45{,}000$ lbs. per sq. in.
$f_c = 96{,}000$ lbs. per sq. in.

Driven rivet diam. Ins.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	Main plate thickness. Ins.
	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	Inside cover plate thickness. Ins.
$\frac{5}{8}$	**10.18** **93.9**	**9.58** **93.5**	**9.18** **93.2**	....	....	....	....	....	....	....	....	....	....	Remarks.
$\frac{11}{16}$	11.49 94.0	10.77 93.6	**10.34** **93.3**	**9.99** **93.1**	....	....	....	....	....	....	....	....	....	The following style of figures indicates failure by crushing throughout the joint and shows a maximum and constant efficiency for the rivet and plate in question:
$\frac{3}{4}$	12.54 94.0	11.75 93.6	**11.50** **93.5**	**11.08** **93.2**	....	....	....	....	....	....	....	....	....	
$\frac{13}{16}$	13.57 94.0	12.72 93.6	**12.72** **93.6**	**12.23** **93.4**	**11.87** **93.2**	....	....	....	....	....	....	....	....	1 2 3 4 5 6 7 8 9 0
$\frac{7}{8}$	14.62 94.0	13.71 93.6	13.86 93.6	**13.41** **93.5**	**12.99** **93.3**	**12.66** **93.1**	....	....	....	....	....	....	....	The following style of figures indicates failure by crushing on the inner row of rivets and single shearing on the outer. With a thickness of inside cover plate greater than that specified in the table, the manner of failure, and hence the pitch and efficiency, will not be changed.
$\frac{15}{16}$	15.66 94.0	14.68 93.6	14.84 93.6	**14.63** **93.6**	**14.14** **93.4**	**13.77** **93.2**	....	....	....	....	....	....	....	
1	16.70 94.0	15.67 93.6	15.84 93.6	**15.88** **93.7**	**15.32** **93.5**	**14.90** **93.3**	**14.55** **93.1**	....	....	....	....	....	....	
$1\frac{1}{16}$	17.75 94.0	16.65 93.6	16.83 93.6	16.97 93.7	**16.55** **93.6**	**16.07** **93.4**	**15.68** **93.3**	....	....	....	....	....	....	**1 2 3 4 5 6 7 8 9 0**
$1\frac{1}{8}$	18.80 94.0	17.62 93.6	17.81 93.6	17.96 93.7	**17.79** **93.7**	**17.25** **93.5**	**16.82** **93.3**	**16.46** **93.2**	....	....	....	....	....	
$1\frac{3}{16}$	19.85 94.0	18.61 93.6	18.81 93.6	18.95 93.7	**19.07** **93.7**	**18.47** **93.6**	**17.98** **93.4**	**17.59** **93.2**	**17.26** **93.1**	....	....	....	....	The outside cover plate thickness so long as it is substantially more than half the main plate thickness does not enter into the pitch calcula-

TABLE XXIII.

MAXIMUM PITCH ON CALKED ROWS OF RIVETS CONSISTENT WITH TIGHTNESS.

Assumed Permissible Deflection of Plate, 0.00035 In.

Driven diam. of rivets. Ins.	Thickness of calked plate. Ins.													Remarks.
	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	13/16	⅞	15/16	1	
⅝	3.02	3.45	3.87	4.26	4.65	5.02	5.38	5.73	6.07	6.41	6.74	7.07	7.39	Working Pressure 100 lbs. per sq. in. Test Pressure 150 lbs. per sq. in.
11/16	3.08	3.51	3.93	4.33	4.71	5.08	5.44	5.79	6.14	6.47	6.80	7.13	7.45	
¾	3.14	3.58	3.99	4.39	4.77	5.14	5.50	5.86	6.20	6.54	6.87	7.19	7.51	
13/16	3.20	3.64	4.05	4.45	4.83	5.20	5.56	5.92	6.26	6.60	6.93	7.25	7.57	
⅞	3.28	3.70	4.12	4.51	4.90	5.27	5.63	5.98	6.32	6.66	6.99	7.32	7.64	
15/16	3.34	3.76	4.18	4.58	4.96	5.33	5.69	6.04	6.39	6.72	7.05	7.38	7.70	
1	3.40	3.83	4.24	4.64	5.02	5.39	5.75	6.11	6.45	6.79	7.12	7.44	7.76	
1 1/16	3.46	3.89	4.30	4.70	5.08	5.45	5.81	6.17	6.51	6.85	7.18	7.50	7.82	
1⅛	3.53	3.95	4.37	4.76	5.15	5.52	5.88	6.23	6.57	6.91	7.24	7.57	7.89	
1 3/16	3.59	4.01	4.43	4.83	5.21	5.58	5.94	6.29	6.64	6.97	7.30	7.63	7.95	
1¼	3.65	4.08	4.49	4.89	5.27	5.64	6.00	6.36	6.70	7.04	7.37	7.69	8.01	
1 5/16	3.71	4.14	4.55	4.95	5.33	5.70	6.06	6.42	6.76	7.10	7.43	7.75	8.07	
1⅜	3.78	4.20	4.62	5.01	5.40	5.77	6.13	6.48	6.82	7.16	7.49	7.82	8.14	
⅝	2.79	3.18	3.55	3.91	4.26	4.59	4.92	5.24	5.55	5.85	6.15	6.45	6.73	Working Pressure 150 lbs. per sq. in. Test Pressure 225 lbs. per sq. in.
11/16	2.85	3.24	3.61	3.97	4.32	4.66	4.98	5.30	5.61	5.92	6.21	6.51	6.80	
¾	2.91	3.30	3.68	4.04	4.38	4.72	5.04	5.36	5.67	5.98	6.28	6.57	6.86	
13/16	2.97	3.37	3.74	4.10	4.45	4.78	5.11	5.43	5.74	6.04	6.34	6.63	6.92	
⅞	3.04	3.43	3.80	4.16	4.51	4.84	5.17	5.49	5.80	7.10	6.40	6.70	6.98	
15/16	3.10	3.49	3.86	4.22	4.57	4.91	5.23	5.55	5.86	7.17	6.46	6.76	7.05	
1	3.16	3.55	3.93	4.29	4.63	4.97	5.29	5.61	5.92	7.23	6.53	6.82	7.11	
1 1/16	3.22	3.62	3.99	4.35	4.70	5.03	5.36	5.68	5.99	7.29	6.59	6.88	7.17	
1⅛	3.29	3.68	4.05	4.41	4.76	5.09	5.42	5.74	6.05	7.35	6.65	6.95	7.23	
1 3/16	3.35	3.74	4.11	4.47	4.82	5.16	5.48	5.80	6.11	7.42	6.71	7.01	7.30	
1¼	3.41	3.80	4.18	4.54	4.88	5.22	5.54	5.86	6.17	7.48	6.78	7.07	7.36	
1 5/16	3.47	3.87	4.24	4.60	4.95	5.28	5.61	5.93	6.24	7.54	6.84	7.13	7.42	
1⅜	3.54	3.93	4.30	4.66	5.01	5.34	5.67	5.99	6.30	7.60	6.90	7.20	7.48	

TABLE XXIII. — *Continued.*

MAXIMUM PITCH ON CALKED ROWS OF RIVETS CONSISTENT WITH TIGHTNESS.

Assumed Permissible Deflection of Plate, 0.00035 In.

Driven diam. of rivets. Ins.	Thickness of calked plate. Ins.													
	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	
5/8	2.64	3.02	3.35	3.68	4.01	4.32	4.62	4.92	5.21	5.49	5.77	6.04	6.31	Working Pressure 200 lbs. per sq. in. Test Pressure 300 lbs. per sq. in.
11/16	2.70	3.08	3.41	3.75	4.07	4.38	4.68	4.98	5.27	5.55	5.83	6.10	6.37	
3/4	2.76	3.14	3.48	3.81	4.13	4.44	4.75	5.04	5.33	5.62	5.89	6.17	6.44	
13/16	2.82	3.20	3.54	3.87	4.19	4.51	4.81	5.10	5.39	5.68	5.96	6.23	6.50	
7/8	2.89	3.27	3.60	3.93	4.26	4.57	4.87	5.17	5.46	5.74	6.02	6.29	6.56	
15/16	2.95	3.33	3.66	4.00	4.32	4.63	4.93	5.23	5.52	5.80	6.08	6.35	6.62	
1	3.01	3.39	3.73	4.06	4.38	4.69	5.00	5.29	5.58	5.87	6.14	6.42	6.69	
1 1/16	3.07	3.45	3.79	4.12	4.44	4.76	5.06	5.35	5.64	5.93	6.21	6.48	6.75	
1 1/8	3.14	3.52	3.85	4.18	4.51	4.82	5.12	5.42	5.71	5.99	6.27	6.54	6.81	
1 3/16	3.20	3.58	3.91	4.25	4.57	4.88	5.18	5.48	5.77	6.05	6.33	6.60	6.87	
1 1/4	3.26	3.64	3.98	4.31	4.63	4.94	5.24	5.54	5.83	6.12	6.39	6.67	6.94	
1 5/16	3.32	3.70	4.04	4.37	4.69	5.01	5.31	5.60	5.89	6.18	6.46	6.73	7.00	
1 3/8	3.39	3.77	4.10	4.43	4.76	5.07	5.37	5.67	5.96	6.24	6.52	6.79	7.06	
5/8	2.53	2.87	3.20	3.52	3.82	4.12	4.41	4.69	4.96	5.23	5.49	5.75	6.00	Working Pressure 250 lbs. per sq. in. Test Pressure 375 lbs. per sq. in.
11/16	2.59	2.94	3.26	3.58	3.89	4.18	4.47	4.75	5.02	5.29	5.55	5.81	6.06	
3/4	2.65	3.00	3.33	3.64	3.95	4.24	4.53	4.81	5.08	5.35	5.61	5.87	6.13	
13/16	2.71	3.06	3.39	3.71	4.01	4.30	4.59	4.87	5.15	5.42	5.68	5.94	6.19	
7/8	2.78	3.12	3.45	3.77	4.07	4.37	4.66	4.94	5.21	5.48	5.74	6.00	5.25	
15/16	2.84	3.19	3.51	3.83	4.14	4.43	4.72	5.00	5.27	5.54	5.80	6.06	6.31	
1	2.90	3.25	3.58	3.89	4.20	4.49	4.78	5.06	5.33	5.60	5.86	6.12	6.38	
1 1/16	2.96	3.31	3.64	3.96	4.26	4.55	4.84	5.12	5.40	5.67	5.93	6.19	6.44	
1 1/8	3.03	3.37	3.70	4.02	4.32	4.62	4.91	5.19	5.46	5.73	5.99	6.25	6.50	
1 3/16	3.09	3.44	3.76	4.08	4.39	4.68	4.97	5.25	5.52	5.79	6.05	6.31	6.56	
1 1/4	3.15	3.50	3.83	4.14	4.45	4.74	5.03	5.31	5.58	5.85	6.11	6.37	6.63	
1 5/16	3.21	3.56	3.89	4.21	4.51	4.80	5.09	5.37	5.65	5.92	6.18	6.44	6.69	
1 3/8	3.28	3.62	3.95	4.27	4.57	4.87	5.16	5.44	5.71	5.98	6.24	6.50	6.75	

CHAPTER IV.

GENERAL PROPORTIONS.

56. Water Consumption. — The standard requirements for boiler rating call for the evaporation of 34.5 lbs. of water from and at 212° F. per horse-power per hour. The total rated water consumption of a boiler of any desired horse-power may thus be found by direct multiplication. Very often it is desired to design a boiler to replace an old one of known evaporative capacity when working under peculiar conditions. It then becomes necessary to reduce the performance of the latter to the terms of the standard rating before the above figures can be applied.

The heat necessary to evaporate one pound of water into steam depends upon the temperature of the incoming feed water, the steam pressure to be maintained and the percentage of moisture in the steam. An actual example will best serve to illustrate the method of calculation. Let it be assumed that an old boiler is to be replaced, the boiler room data showing that, under average conditions, 3500 lbs. of feed water at 150° F. are evaporated per hour into steam at 135 lbs. per sq. in. gage pressure with 2 per cent of moisture or priming.

The temperature corresponding to an absolute pressure of 149.7 lbs. per sq. in. is 358.3° F. By reference to the heats of the liquid at the temperatures of the steam and feed water, it is evident that

$$329.9 - 118.0 = 211.9 \text{ B.T.U.}$$

are necessary to raise each pound of feed water to the temperature of the steam. To vaporize 98 per cent of the feed water at 358.3° F. requires the expenditure of

$$0.98 \times 863.2 = 845.9 \text{ B.T.U.}$$

The total amount of heat required therefore per pound of water evaporated is

$$211.9 + 845.9 = 1057.8 \text{ B.T.U.}$$

In comparison with this it should be noted that the evaporation of one pound of water under the standard conditions of boiler rating, namely from feed water at 212° F. to dry steam at the same temperature, calls for the expenditure of 969.7 B.T.U. Therefore, one boiler horse-power under the conditions of the problem is equivalent to the evaporation of

$$\frac{969.7}{1057.8} \times 34.5 = 31.61 \text{ lbs. of water per hr.}$$

The horse-power of the boiler to be replaced is, therefore,

$$\frac{3500}{31.61} = 110.8$$

and the design of the new one should proceed with this figure as a basis.

In order to facilitate the calculation of standard ratings under a wide variety of conditions, the following table is inserted. When the total approximate consumption of feed water per hour is known, the horse-power of the boiler may be found by dividing by the equivalent evaporation as expressed in Table XXIV under the actual operating conditions. Conversely the expected actual water consumption of a boiler of given horse-power may be found by multiplying by the tabular quantity corresponding to the conditions of feed temperature and steam pressure.

57. Coal Consumption.—While the designer is rarely expected to specify the exact type of fuel for which the boiler shall be used, it is necessary for him to have a definite idea of the essential characteristics of fuels in general. The number of pounds of water evaporated per pound of combustible varies widely with the kind of fuel, the rate of combustion and the type of boiler. With uniform draft conditions the character of the fuel influences most the main features of the design. The kinds of fuel most generally used at present for purposes of steam generation are as follows:

(*a*) Coal
(*b*) Coke
(*c*) Crude Oil
(*d*) Natural Gas
(*e*) Wood
(*f*) Waste Products

Coal is classed in the United States according to the amount of volatile matter which it contains, in the following manner:

TABLE XXIV.

ACTUAL EVAPORATION.

Lbs. of Water Corresponding to One Boiler Horse-Power.

(34.5 Lbs. per Hour from and at 212° F.)

For Various Feed-water Temperatures and Steam Pressures.

Priming assumed 2 per cent.

emperature of feed water. Degrees F.	Gage Pressure. Lbs. per sq. in.								
	50	75	100	125	150	175	200	225	250
32	28.83	28.71	28.56	28.46	28.39	28.34	28.30	28.25	28.19
35	28.92	28.78	28.64	28.54	28.46	28.42	28.36	28.32	28.28
40	29.03	28.92	28.76	28.68	28.59	28.54	28.49	28.44	28.39
45	29.17	29.03	28.89	28.80	28.71	28.66	28.61	28.56	28.52
50	29.29	29.17	29.01	28.94	28.83	28.78	28.74	28.68	28.64
55	29.43	29.29	29.14	29.06	28.96	28.92	28.86	28.80	28.76
60	29.54	29.43	29.27	29.18	29.09	29.03	28.98	28.94	28.89
65	29.69	29.54	29.40	29.32	29.21	29.17	29.12	29.06	29.01
70	29.82	29.69	29.52	29.47	29.34	29.29	29.24	29.18	29.14
75	29.95	29.82	29.66	29.57	29.47	29.43	29.37	29.32	29.27
80	30.08	29.95	29.80	29.71	29.60	29.54	29.49	29.44	29.40
85	30.23	30.08	29.93	29.84	29.74	29.69	29.63	29.57	29.52
90	30.35	30.23	30.06	29.97	29.86	29.82	29.77	29.71	29.66
95	30.49	30.35	30.20	30.11	30.00	29.95	29.90	29.84	29.80
100	30.63	30.49	30.33	30.23	30.14	30.08	30.03	29.97	29.93
105	30.77	30.63	30.46	30.38	30.28	30.23	30.17	30.11	30.06
110	30.91	30.77	30.61	30.49	30.41	30.35	30.31	30.25	30.20
115	31.12	30.91	30.74	30.63	30.54	30.49	30.43	30.38	30.33
120	31.20	31.06	30.89	30.77	30.69	30.63	30.57	30.52	30.46
125	31.34	31.20	31.02	30.91	30.82	30.77	30.71	30.66	30.61
130	31.50	31.34	31.18	31.06	30.97	30.91	30.85	30.80	30.74
135	31.64	31.50	31.32	31.20	31.11	31.06	30.99	30.94	30.89
140	31.80	31.64	31.47	31.34	31.26	31.20	31.14	31.09	31.02
145	31.96	31.80	31.61	31.50	31.41	31.34	31.29	31.23	31.18
150	32.10	31.96	31.77	31.64	31.56	31.50	31.44	31.37	31.47
155	32.26	32.10	31.96	31.80	31.71	31.64	31.59	31.53	31.61
160	32.42	32.26	32.07	31.96	31.85	31.80	31.74	31.67	31.77
165	32.65	32.42	32.23	32.10	32.01	31.96	31.89	31.83	31.32
170	32.73	32.58	32.39	32.26	32.17	32.10	32.04	31.99	31.93
175	32.91	32.73	32.55	32.42	32.32	32.26	32.21	32.14	32.07
180	33.06	32.91	32.71	32.58	32.48	32.42	32.35	32.29	32.23
185	33.23	33.06	32.87	32.76	32.63	32.58	32.52	32.45	32.39
190	33.39	33.23	33.03	32.94	32.79	32.73	32.67	32.60	32.55
195	33.56	33.39	33.20	33.09	32.96	32.91	32.84	32.76	32.71
200	33.73	33.55	33.36	33.26	33.16	33.06	32.99	32.94	32.91
205	33.90	33.72	33.53	33.42	33.32	33.23	33.16	33.09	33.06
210	34.06	33.89	33.70	33.57	33.47	33.39	33.32	33.26	33.23
212	34.14	33.96	33.78	33.64	33.55	33.46	33.39	33.32	33.29

CLASSIFICATION OF COAL
in regard to
VOLATILE MATTER.
U. S. Geol. Survey.

ANTHRACITE: When volatile matter is not more than 7½ per cent of the total combustible.

SEMI-ANTHRACITE: When volatile matter is from 7½ to 12½ per cent of the total combustible.

SEMI-BITUMINOUS: When volatile matter is from 12½ to 25 per cent of the total combustible.

BITUMINOUS: When volatile matter is from 25 to 50 per cent of the total combustible.

LIGNITE: When volatile matter is over 50 per cent of the total combustible.

This chemical classification is roughly followed by a geographical one as well, since eastern Pennsylvania coals are quite generally anthracite while those in the central portion of the Allegheny-Atlantic valley are semi-bituminous in character. The coals of western Pennsylvania, together with those of the southern states and Mississippi valley, are of the bituminous variety, the percentage of volatile matter increasing with but few exceptions in the far West and Southwest. In the regions of the Northwest many deposits of lignitic coals are found.

TABLE XXV.
ANTHRACITE COAL SIZES.

Name.	Standard square mesh. Ins.	
	Through.	Over.
Broken	4	2¾
Egg	2¾	2
Stove	2	1⅜
Chestnut	1⅜	¾
Pea	¾	½
No. 1 buckwheat	½	¼
No. 2 buckwheat or rice	¼	⅛
No. 3 buckwheat or barley	⅛	$\frac{1}{16}$

Anthracite coal is ordinarily placed upon the market under the sizes and names given in Table XXV.

Coke is the product of coal, generally bituminous in character, when subjected to complete devolatilization in the gas house retort or coke oven. It therefore contains only traces of the hydrocarbons and consists essentially of pure carbon with certain admixed impurities. The latter, especially in the case of sulphur, may cause considerable trouble by fouling the grate and furnace walls, when the coke is burned under conditions of forced draft. Coke is rarely used as a steam fuel unless external conditions render it necessary. Smoke prevention in the residential districts of cities and the comfort of railway passengers in long underground tunnels generally make the use of coke imperative.

The use of oil as a fuel in the furnaces of steam boilers has recently attained proportions of great importance. Especially is this true in localities where oil is plenty and coal or other fuel comparatively scarce. Oil fuel is of two kinds, namely, the crude petroleum as it comes from the wells, and petroleum from which the lighter distillates have been removed. While for a given weight the former contains fewer heat units than the latter, the crude oil is of much greater specific gravity, and on the basis of calorific value for the volume occupied, has the advantage. One pound of crude oil contains between 18,000 and 19,000 British Thermal Units and is therefore the equivalent of 1.37 pounds of bituminous coal or 1.67 pounds of anthracite in so far as heat units are concerned. The equivalent evaporative power of oil is, however, greater than these figures would indicate since the ease with which the furnace conditions may be manipulated permits of a more perfect combustion.

Oil burners may use air or steam as the atomizing agent and the mixing may take place within or without the burner. When it is not expedient to use either air or steam a mechanical atomizer may be employed. The latter consists of a small orifice loosely closed by a plug. Upon the surface of the latter are helical grooves from which the oil issues at a high velocity, forming a rotary spray. Since the use of air causes additional complications in the plant, air burners are employed, as a rule, only where high temperatures are required or where the effect of the steam jet would be injurious. In metallurgical and chemical processes the presence of so much moisture in the flame is injurious to the products. Air burners are more liable to clog with carbon than are steam burners. In general the arrangement of the furnace has more to do with the efficiency of the plant than has the type and form of burner used. For the atomizing of one pound of oil, from one-quarter to one-half pound of steam is required, or from one and one-half to three per cent of the total steam generated. Since furnace temperatures in oil burners reach from 2000° to 2400° F., and in extreme cases 2800° F., it is necessary to protect the boiler shell from the direct action of the flames. This is accomplished by baffle plates and linings of refractory bricks.

The use of natural gas under steam boilers is restricted to the few localities where the proximity of gas wells and their uniformity of flow make it possible. As a cheap and cleanly fuel, easily controlled and adapted to many conditions, it has great advantages.

Owing to its scarcity and high cost, wood as a permanent fuel for steam power purposes has been largely superseded. It is only in frontier sections of the country that the supply is plentiful enough to warrant its use, and generally in such cases the wood burned occurs as a by-product of processes such as those of lumbering and paper mill operation. The hazard from flying sparks is great and the cost of transportation and handling is excessive.

Many industries involve the use of materials which, after having performed their natural purposes, may be burned as waste products. Sugar cane from which the juice has been expressed, commonly known as bagasse, tan bark which has yielded its principle to the tan liquor, as well as chips, sawdust, leather parings and paper stock waste fall under this class of fuel. While many materials formerly classed as waste products are now turned to excellent account in manufactures, there still remains much refuse which alone can be burned. The design of conveyors and furnaces for such products is largely a matter of experiment and must be considered as a special problem.

In order to facilitate the selection of a reasonable figure for the number of pounds of water evaporated per pound of fuel, the following table has been compiled from the results of a wide range of tests. While the rate of consumption has some effect upon the evaporation, the kind of fuel is much more directly concerned. To discriminate between the evaporative efficiencies of different types of boilers is a difficult and unsatisfactory task. The tight-

TABLE XXVII.

FACTORS OF EVAPORATION.

For Various Feed-water Temperatures and Steam Pressures as compared with Evaporation from and at 212° F.

Priming assumed 2 per cent.

Temperature of feed water. Degrees F.	Gage Pressure. Lbs. per sq. in.								
	50	75	100	125	150	175	200	225	250
32	1.196	1.201	1.208	1.212	1.215	1.216	1.219	1.221	1.224
35	1.193	1.199	1.205	1.209	1.212	1.214	1.216	1.218	1.220
40	1.188	1.193	1.199	1.203	1.207	1.209	1.211	1.214	1.215
45	1.182	1.188	1.194	1.198	1.201	1.203	1.206	1.208	1.210
50	1.178	1.182	1.190	1.192	1.196	1.199	1.200	1.203	1.205
55	1.172	1.178	1.181	1.187	1.191	1.193	1.195	1.198	1.199
60	1.168	1.172	1.179	1.182	1.186	1.188	1.191	1.192	1.194
65	1.162	1.168	1.173	1.177	1.181	1.182	1.185	1.187	1.190
70	1.157	1.162	1.169	1.170	1.176	1.178	1.180	1.182	1.184
75	1.152	1.157	1.163	1.166	1.170	1.172	1.174	1.177	1.179
80	1.147	1.152	1.158	1.161	1.165	1.168	1.169	1.171	1.173
85	1.141	1.147	1.153	1.156	1.160	1.162	1.164	1.166	1.169
90	1.137	1.141	1.148	1.151	1.155	1.157	1.159	1.161	1.163
95	1.132	1.137	1.142	1.146	1.150	1.152	1.153	1.156	1.158
100	1.127	1.132	1.137	1.141	1.145	1.147	1.149	1.151	1.153
105	1.121	1.127	1.132	1.136	1.140	1.141	1.144	1.146	1.148
110	1.116	1.121	1.127	1.132	1.134	1.137	1.139	1.140	1.142
115	1.108	1.116	1.122	1.127	1.129	1.132	1.133	1.136	1.137
120	1.106	1.111	1.117	1.121	1.124	1.127	1.128	1.131	1.132
125	1.101	1.106	1.113	1.116	1.119	1.121	1.123	1.126	1.127
130	1.095	1.101	1.107	1.111	1.114	1.116	1.118	1.120	1.122
135	1.090	1.095	1.102	1.106	1.109	1.111	1.113	1.115	1.117
140	1.085	1.090	1.096	1.101	1.104	1.106	1.108	1.110	1.112
145	1.080	1.085	1.091	1.095	1.098	1.101	1.103	1.105	1.107
150	1.074	1.080	1.086	1.090	1.093	1.095	1.097	1.100	1.102
155	1.069	1.074	1.081	1.085	1.088	1.090	1.092	1.094	1.096
160	1.064	1.069	1.075	1.080	1.082	1.085	1.087	1.089	1.091
165	1.056	1.064	1.070	1.075	1.077	1.080	1.081	1.084	1.086
170	1.054	1.059	1.065	1.069	1.073	1.074	1.076	1.078	1.081
175	1.047	1.054	1.060	1.064	1.068	1.069	1.071	1.074	1.075
180	1.043	1.048	1.054	1.059	1.062	1.064	1.066	1.069	1.070
185	1.039	1.043	1.050	1.053	1.057	1.059	1.061	1.063	1.065
190	1.033	1.039	1.044	1.047	1.052	1.054	1.056	1.058	1.060
195	1.028	1.033	1.040	1.042	1.047	1.048	1.051	1.053	1.054
200	1.023	1.028	1.034	1.037	1.040	1.043	1.046	1.047	1.048
205	1.017	1.023	1.029	1.032	1.035	1.039	1.040	1.042	1.043
210	1.013	1.018	1.023	1.028	1.030	1.033	1.035	1.037	1.039
212	1.010	1.016	1.021	1.025	1.028	1.031	1.033	1.035	1.036

For example, let it be desired to find the expected actual evaporation per pound of mine-run bituminous coal for a horizontal return tubular boiler, using feed water at a temperature of 130° F. and carrying steam at a gage pressure of 125 lbs. per sq. in. As before, let the priming be assumed 2 per cent. The temperature corresponding to an absolute pressure of 139.7 lbs. per sq. in. is 352.9° F. Consulting the heats of the liquid at the temperatures of the steam and feed water, it is evident that

$$324.3 - 98.0 = 226.3 \text{ B.T.U.}$$

per pound will be necessary to raise the feed water to the steam temperature. To vaporize 98 per cent of the feed water into steam at an absolute pressure of 139.7 lbs. per sq. in. will require

$$0.98 \times 867.5 = 850.2 \text{ B.T.U.}$$

per pound. The total heat necessary to evaporate water under these actual conditions will, therefore, be

$$226.3 + 850.2 = 1076.5 \text{ B.T.U.}$$

The ratio of this to the standard heat of vaporization from and at 212° F.

$$\frac{1076.5}{969.7} = 11.11$$

is the factor of evaporation. Referring to Table XXVI the standard evaporation for the coal assumed above may be taken as 9.5 lbs. per pound of coal. The actual evaporation will then be

$$\frac{9.5}{11.11} = 8.555 \text{ lbs.}$$

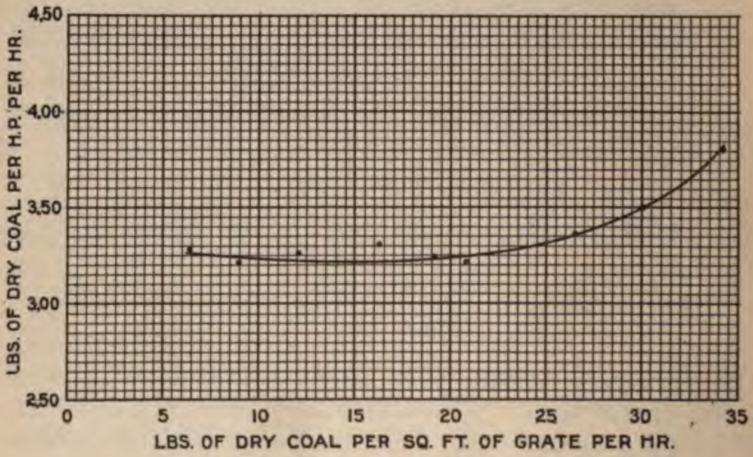
Conversely, if an old boiler has shown an actual evaporative capacity of 10 lbs. of water per pound of coal under the peculiar conditions given above, its reduced evaporation will be

$$10 \times 11.11 = 11.11 \text{ lbs.}$$

per pound of coal, and the design of a new boiler to replace the old one should proceed from the latter figure as a basis.

58. Grate Surface. — After having calculated the total rated water consumption and the gross amount of fuel necessary for a given design, it next remains to apportion the grate surface. For every kind of fuel there is a certain rate of combustion which yields the greatest evaporative economy. When a very low draft pressure is used the oxygen of the air does not penetrate the fuel-bed sufficiently to produce complete combustion. A large percentage of fuel drops through the grate also. On the other hand, a very strong draft tends to carry large quantities of fine unconsumed fuel over the bridge wall and up the stack.

In 1896 Mr. J. M. Whitham made a series of tests upon a 100 horse-power horizontal multitubular boiler for the Philadelphia Traction Co., to determine the value of certain auxiliary furnace apparatus in gaining a more perfect heat absorption. The tests are reported in Vol. XVII, Transactions of the A.S.M.E. As a basis for this work he made a very complete series of standard boiler tests with varying draft conditions. In the plot, Fig. 105, is shown the variation in dry coal consumed per horse-power per hour and the rate of combustion. For rates above 20 lbs. of coal per sq. ft. of grate surface per hour there is a rapid diminution in efficiency. At the left end of the curve there is also a loss though not so pronounced. At the point of best efficiency the boiler was working under an overload comprising 170 per cent of its rated horse-power. The fuel was a uniform grade, run-of-mine Pennsylvania bituminous coal.



EFFICIENCY OF COMBUSTION
BITUMINOUS COAL
Fig. 105.

As a basis for the calculation of the grate area, therefore, a rate of combustion must be selected which gives a large margin for economy. Table XXVIII gives the results of many tests under varying conditions and may be taken to represent good average practice in apportioning the grate surface. The A.S.M.E. boiler rating states that the standard conditions include a draft in the flue just beyond the boiler of at least one-half inch water column. The quantities given in Table XXVIII can easily be attained under this draft pressure.

TABLE XXVIII.

AVERAGE RATES OF COMBUSTION.

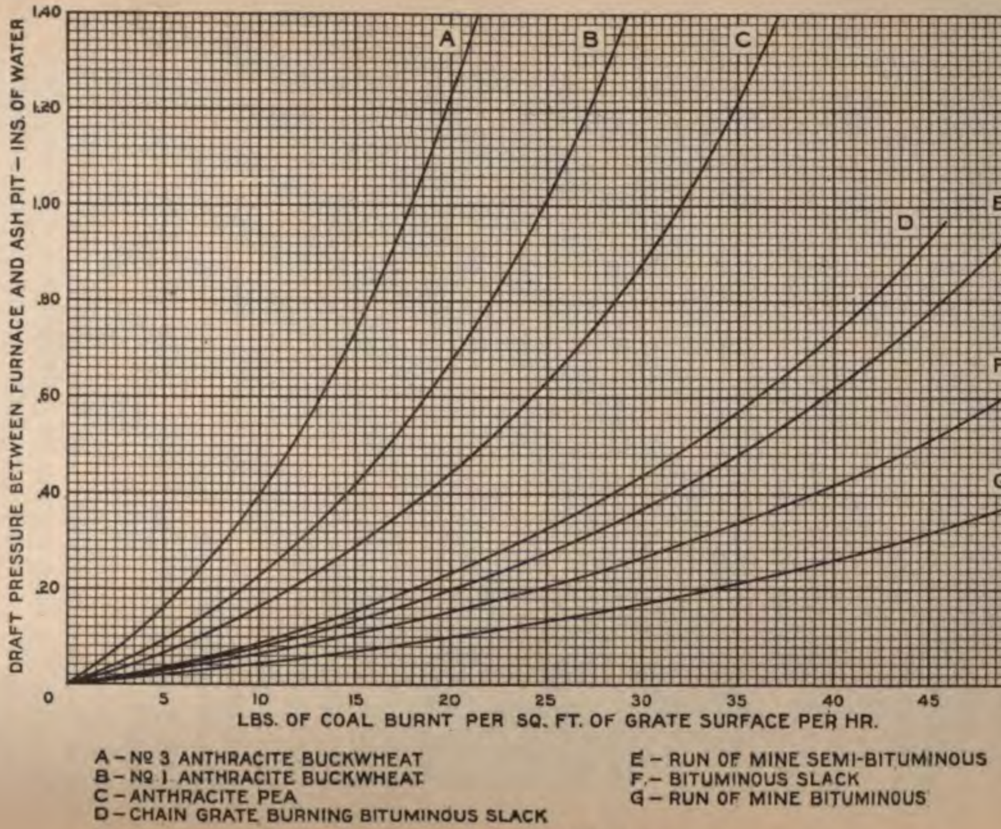
Lbs. per Sq. Ft. of Grate Surface per Hour.

Draft, ½ in. Water Column.

Fuel used.	Stationary grate.	Fuel used.	Stationary grate.
Anthracite:		Bituminous:	
Slack	9	Slack	14
Buckwheat	12	Eastern, run-of-mine	18
Pea	14	Western, run-of-mine	20
Semi-bituminous:		Lignite	12
Run-of-mine	20		

It is frequently the case that a new boiler is to be designed to replace an old one, the coal consumption of the latter having taken place under artificial or unusual conditions. It is then necessary to select constants which shall harmonize with the conditions of the previous design. The plot of Fig. 106 is inserted by special permission of the Babcock & Wilcox Co. It represents the coal consumption per sq. ft. of grate surface with a wide variety of draft pressures in the ash-pit.

The intensity of natural chimney draft is greatest at the base of the stack and rapidly diminishes along the pathway of the gases toward the ash-pit. This diminution is due to the frictional resistance offered by the flues, smoke passages and bed of fuel. With the ash-pit doors open as they should be, giving free access to the atmosphere, the draft pressure in the ash-pit just below the grate is negligible in amount. The greatest loss in the draft pressure is ordinarily due to the resistance in passing through the fuel upon the grate. The other losses occurring are those due to peculiarities in the setting, the turns in the uptake and the friction of the gases on the sides of the tubes and passages. Losses of the

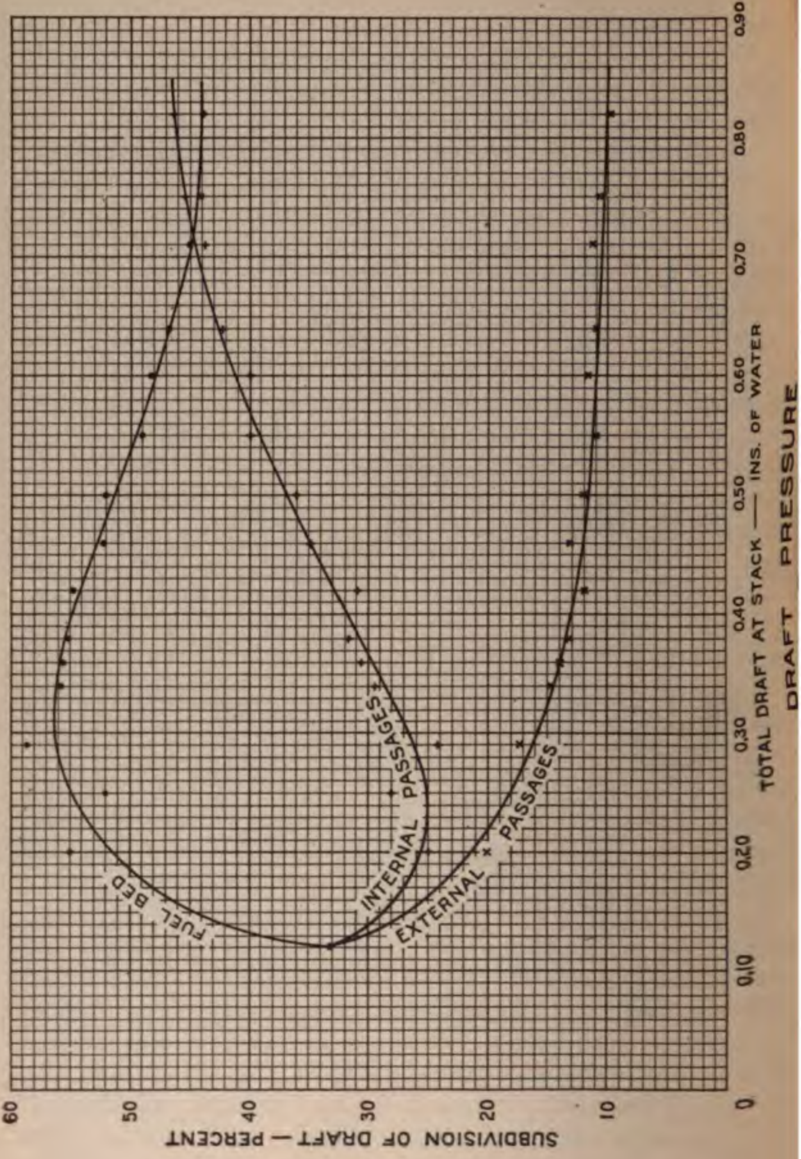


DRAFT AND COMBUSTION

Fig. 106.

latter character vary widely with the type of boiler and the arrangement of the setting.

There are three different stations at which draft readings are ordinarily taken: (*a*) in the furnace just above the fire; (*b*) at the uptake damper where the hot gases leave the boiler setting; (*c*) at the base of the stack. In boilers having very complex settings other points of observation may be established along the line of the gases. Many experiments have been made to determine the probable sub-division of a given stack pressure among the various points of loss mentioned above. The losses in the fuel-bed and tubes are not constant for all gas velocities but vary widely with the coal consumption. This fact is attested by data furnished in connection with the tests reported by Mr. J. M. Whitham to which reference was made above. The return tubular boiler used in his tests was fitted with a return flue or pass along the top of the boiler shell. Draft pressures were measured in the furnace just over the fire, at the front end of the tubes in the smoke-box extension and at the uptake damper near the rear end of the top pass. The losses in the tubes and fuel-bed varied widely, while that in the top pass was a small and nearly constant one. The top pass is not often applied to multitubular boilers at present. From its connection at the front end of the setting it resembles somewhat the usual uptake, except that the latter is generally made of bare sheet iron. While these tests strictly apply to the above type of boiler only, they were carried out with such care and with so wide a range of combustion that they shed valuable light on draft performance in general. The plot in Fig. 107 is constructed from numerical data taken from Mr. Whitham's tables of draft and coal consumption. With the total draft measured in inches of water as abscissæ the per cent loss at the fuel-bed, the internal gas passages and the external or top passage, respectively, are plotted as ordinates. Three very distinct lines result. The first shows a gradually falling percentage of loss at the fuel-bed, the second a slowly increasing per cent loss in the internal passages and the third a diminishing but nearly constant loss in the external passage. These results appear to be well founded and afford a very reasonable method of apportioning the total chimney draft among the three chief factors which absorb it. With natural draft showing an intensity of 0.50 in. of water at the stack, for example, it is reasonable to expect that the fuel-bed will absorb about 52 per



DRAFT PRESSURE

cent or 0.26 in. Referring with this figure to the plot of Fig. 106 and the curve for Pennsylvania bituminous coal, the consumption as indicated is 22.0 lbs. per sq. ft. of grate surface per hour. This figure is precisely the consumption that Mr. Whitham's tests showed under similar conditions.

While, strictly speaking, the above results are alone applicable to the type of boiler and fuel used in the tests, they afford at the same time interesting suggestions in regard to draft performance in general. Doubtless with different sizes of fuel and boiler tubes the percentage of sub-division would vary widely. It is not claimed that these experiments furnish exact data for all possible cases, but, as a reasonable approximation, it is believed that the curves of Fig. 107 afford a fair means of estimating the portion of the total draft which is available for combustion at the grate.

In designing boilers, therefore, it is well first to measure or estimate the total available draft. Then by means of the plots in the two preceding figures, determine the percentage at the fuel-bed and the corresponding consumption of coal per square foot of grate. From this the necessary grate surface for a given gross coal consumption may be determined. The practical size of grate to use depends upon the type of boiler and is discussed in Art. 69, page 295, of the present chapter.

When extra long uptakes are used or economizer apparatus is installed between the boiler and stack it is customary to make a gross allowance for the draft necessary to take account of such appliances. For uptakes 0.10 in. of water is allowed for each 100 ft. of straight run and 0.05 in. of water for each right-angle turn. For well installed economizers a gross allowance of 0.30 in. of water is customary.

59. Boiler Tubes. — The best grade of charcoal wrought iron was formerly used for lap-welded boiler tubes. This material was virtually superseded by welded soft Bessemer steel about ten years ago. Since 1910 seamless open-hearth steel, drawn either hot or cold, has become the material preëminent for boiler tubes. The steel ingot pierced and rolled is submitted to successive drawing operations, with the result that a tough reliable material is produced having neither seam nor weld throughout its structure. Boiler tubes are sized according to the dimensions given in Table XXIX. The outside diameter is the nominal diameter of the tube.

TABLE XXIX.

PROPORTIONS OF STANDARD BOILER TUBES.

National Tube Co.

Nominal diam. Actual external diam.	Actual internal diam.	Thickness.		External circumference.		Internal circumference.	
Ins.	Ins.	Ins.	B.W.G.	Ins.	Ft.	Ins.	Ft.
1¾	1.560	0.095	13	5.498	0.4582	4.901	0.4084
2	1.810	0.095	13	6.283	0.5236	5.686	0.4738
2¼	2.060	0.095	13	7.069	0.5891	6.472	0.5393
2½	2.282	0.109	12	7.854	0.6545	7.169	0.5974
2¾	2.532	0.109	12	8.639	0.7199	7.955	0.6629
3	2.782	0.109	12	9.425	0.7854	8.740	0.7283
3¼	3.010	0.120	11	10.210	0.8508	9.456	0.7880
3½	3.260	0.120	11	10.996	0.9163	10.242	0.8535
3¾	3.510	0.120	11	11.781	0.9818	11.027	0.9189
4	3.732	0.134	10	12.566	1.0472	11.724	0.9770
4½	4.232	0.134	10	14.137	1.1781	13.295	1.1079
5	4.704	0.148	9	15.708	1.3090	14.778	1.2315
6	5.670	0.165	8	18.850	1.5708	17.813	1.4844

Nominal diam. Actual external diam.	External transverse area.		Internal transverse area.		Length of tube per sq. ft. of internal heating surface.	Wt. per ft.
Ins.	Sq. ins.	Sq. ft.	Sq. ins.	Sq. ft.	Ft.	Lbs.
1¾	2.405	0.0167	1.911	0.0133	2.448	1.679
2	3.142	0.0218	2.573	0.0179	2.110	1.932
2¼	3.976	0.0276	3.333	0.0231	1.854	2.186
2½	4.909	0.0341	4.090	0.0284	1.674	2.783
2¾	5.940	0.0412	5.035	0.0350	1.508	3.074
3	7.069	0.0491	6.079	0.0422	1.373	3.365
3¼	8.296	0.0576	7.116	0.0494	1.269	4.011
3½	9.621	0.0668	8.347	0.0580	1.172	4.331
3¾	11.045	0.0767	9.676	0.0672	1.088	4.652
4	12.566	0.0873	10.939	0.0760	1.024	5.532
4½	15.904	0.1104	14.066	0.0977	0.903	6.248
5	19.635	0.1364	17.379	0.1207	0.812	7.669
6	28.274	0.1963	25.250	0.1750	0.674	10.282

The thickness of the walls of boiler tubes is usually made to conform to the numbers of the Birmingham Wire Gage. A thickness of tube can be obtained for special purposes, one, two, three, or four gages thicker than the standard. The wire gages and sizes concerned in boiler tube thicknesses are given in Table XXX.

TABLE XXX.

BIRMINGHAM WIRE GAGE NUMBERS and CORRESPONDING BOILER TUBE THICKNESSES.

Actual thickness of tube.	Birm. Wire Gage.	Actual thickness of tube.	Birm. Wire Gage.
Ins.	No.	Ins.	No.
0.095	13	0.165	8
0.109	12	0.180	7
0.120	11	0.203	6
0.134	10	0.220	5
0.148	9	0.238	4

For locomotive practice tubes much heavier than those tabulated above are necessary. Such tubes are made both from seamless and lap-welded steel. Their external diameters vary from 1¼ ins. to 3 ins. and each size is rolled with thicknesses of wall ranging through some nine different wire gages. The actual dimensions can be procured from manufacturers' handbooks.

There are two methods of specifying the limiting size of tube for a given case. The smaller the tubes the greater the total heating surface for the volume occupied in the boiler. Small tubes easily clog with soot and choke the draft. Large tubes are wasteful of room and fail in efficient heat absorption.

A common rule is to allow one inch of nominal diameter for each 4 ft. of length when bituminous coal is used, and for each 5 ft. of length when the fuel is anthracite, the boiler being operated under natural draft. With artificial draft, such as is generally used in marine and locomotive practice, tubes of much less diameter than those specified by the above rules can be used.

A second method consists of prescribing an arbitrary limit for the ratio of length to nominal diameter which must not be exceeded. The value of this ratio is 48 and 60 respectively in the two cases given in the previous rule. For vertical boilers where the circuit of the flames is short and direct, the ratio is generally placed at 70. For horizontal multitubular boilers a ratio of 60 to 65 is found to work well unless the coal is of a very sooty character. In marine type boilers where the limited grate room demands a high rate of combustion combined with efficient heating surface, the ratio is often lowered to 40 or 45.

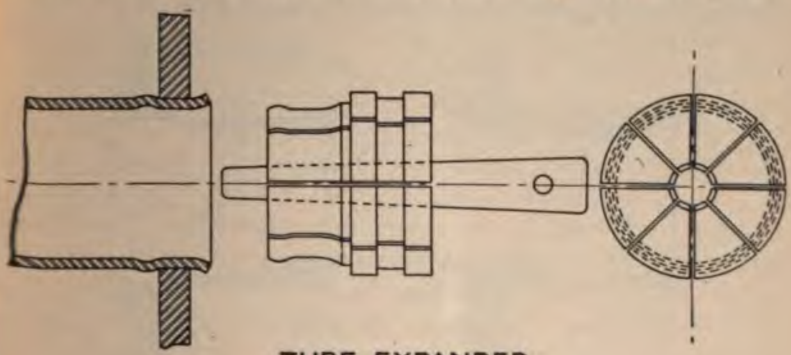
The number of tubes is determined from the practical relation of their total internal transverse area to the grate area. It has been found by experiment that the calorific or smoke area should range from one-sixth to one-eighth the grate area. In vertical boilers where the entrance to the tubes is close to the grate and the contraction in area is sudden, the smoke area should constitute a fairly large percentage of that of the grate. In such cases the smoke area is generally made one-sixth the grate area. Where the pathway of the gases from grate to tubes has considerable length, as in horizontal multitubular boilers, a ratio of one-seventh or one-eighth is found to give good results. The actual number of tubes to be used must be finally determined by the possibility of placing them satisfactorily in the proposed tube sheet.

Commercial boiler tubes are usually trimmed accurately at the ends and vary in length by integral half feet. The distance between the tube sheets must, therefore, be made to conform to the over-all length of the tube.

The fastening of the tubes into the sheet at their ends is a matter of great importance in the construction of pressure vessels. For ordinary methods the hole in the tube sheet should first be accurately bored to the exact outside diameter of the tube, and the sharp corners rounded on both the inner and outer surfaces to a radius of about $\frac{1}{16}$ in. The tube end should project beyond the tube sheet surface a distance of $\frac{1}{4}$ in. With tubes having plain ends the two following methods are employed. In the first a sectional tube expander is driven forcibly into the open end and distorts the tube as shown in Fig. 108. The central taper pin separates the segments of the expander and shapes the ribs both on the inside and outside of the tube sheet. This is known as the Prosser method. During the process the expander is frequently withdrawn and turned about to secure uniformity.

Second, an expanding roller mandrel is introduced into the open end of the tube and rotated, swaging the tube firmly against the tube sheet. The latter is called the Dudgeon method. The projecting end is left by this process as shown in Fig. 109, but may be spread by use of a conical tool to the flared shape shown in Fig. 108.

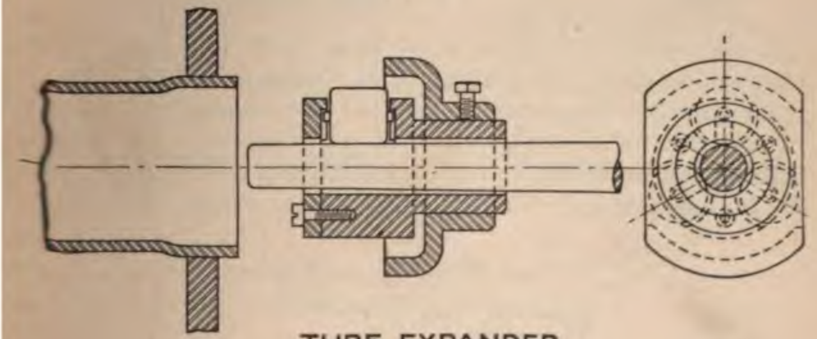
The metal of the tube is not seriously injured by either of the above processes when skilfully performed, and tubes thus treated form efficient stays for binding together the two tube sheets. The action of the fire, however, on the projecting edge of the tube is very severe and soon burns it away. Hence the tubes are almost always subjected to the further operation of beading. This leaves the finished work in the form shown in Fig. 110 for the two methods described above. The form of beading tool used is shown in Fig. 111. It is actuated either pneumatically or by hand. The tube material may become more or less granular under the operation of beading, in which case its holding power as a stay is seriously impaired. In a series of tests in the laboratory of the Massachusetts Institute of Technology made in April, 1910, Mr. C. D. Carey found the holding power of 2½ in. tubes when beaded about 10 per cent less than when simply flared. The manner of



TUBE EXPANDER

PROSSER METHOD

Fig. 108.

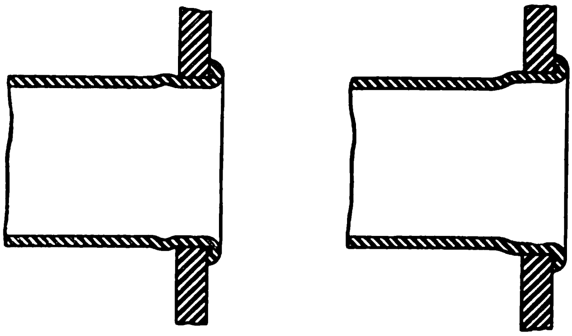


TUBE EXPANDER

DUDGEON METHOD

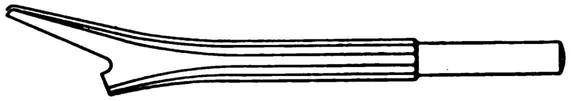
Fig. 109.

failure gave evidence of granulation produced by excessive beading. When the operation is carefully performed upon metal of good ductility, the holding powers of beaded and flared tubes are sensibly equal. The durability of beaded tubes in resisting the action of the fire is much greater than that of the flared type.



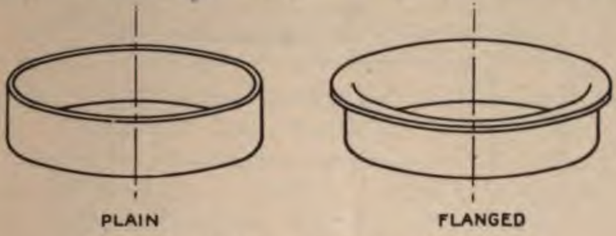
BEADED TUBES
FIG. 110.

A third method of fastening the ends to the tube sheet is by electrical welding. After having been swaged to a tight fit by a roller expander, the protruding tube end is surrounded with a bead of soft steel electrically fused to place. This process is widely used in modern locomotives at the fire box end of the tubes where the intense heat and vibration tend to cause serious difficulty.

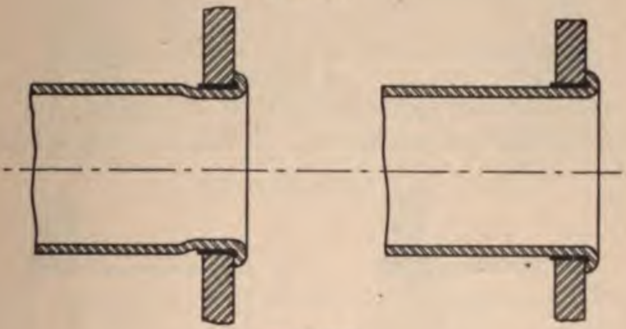


BEADING TOOL
FIG. 111.

Where the effects of expansion are very severe upon the tube ends, ferrules are used. These consist of thin rings, either plain or beaded, made of soft copper, brass or steel, as shown in Fig. 112. There are two methods of application quite different from one another. The first is illustrated in Fig. 113. The tube hole is accurately bored to a diameter slightly larger than that of the outside of the tube, as shown at the right. Soft copper ferrules, generally of the beaded pattern and not over $\frac{1}{16}$ in. thick, are then inserted in the holes and lightly rolled in place. Lastly the tube is introduced from the left and expanded and beaded securely as previously described. Such joints are generally made at the fire box end of the tube only. If both ends are to be ferruled a similar process may be employed. In locomotive practice the tube hole is sometimes bored of the same diameter as the outside of the tube. After rolling in the ferrule, the tube slightly swaged down at the end is inserted and expanded securely to place. This gives the fire box joint shown in Fig. 113 at the left.



COPPER FERRULES
FOR
BOILER TUBES
FIG. 112.



TUBES
SET WITH
FERRULES
OUTSIDE METHOD
FIG. 113.

The usual sizes of soft copper ferrules for this purpose are given in Table XXXI.

TABLE XXXI.

STANDARD SEAMLESS SOFT COPPER FERRULES.

For Use upon Boiler Tubes.

Procurable thickness of stock for all sizes $\frac{1}{32}$, $\frac{3}{64}$, $\frac{1}{16}$, $\frac{5}{64}$, $\frac{3}{32}$, $\frac{7}{64}$, $\frac{1}{8}$ in.

Inside diam.	Widths of ferrule.					
Ins.	Ins.					
1	$\frac{1}{2}$	$\frac{5}{8}$	..	..	..	..
$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	..	..	..	..
$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	..	..	..
$1\frac{3}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	..	..	..
2	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	..
$2\frac{1}{4}$	..	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	..
$2\frac{1}{2}$	..	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	..
$2\frac{3}{4}$	..	..	$\frac{3}{4}$	$\frac{7}{8}$	1	..
3	..	..	$\frac{3}{4}$	$\frac{7}{8}$	1	..
$3\frac{1}{4}$	..	..	$\frac{3}{4}$	$\frac{7}{8}$	1	..
$3\frac{1}{2}$	..	..	$\frac{3}{4}$	$\frac{7}{8}$	1	..
4	..	..	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$

Fig. 114 illustrates the second method of using ferrules. When the tube material is especially weak or thin and is therefore liable to spring back and cause leakage, strong ferrules of hard brass or steel are driven into the inside of the tubes at the ends. By this

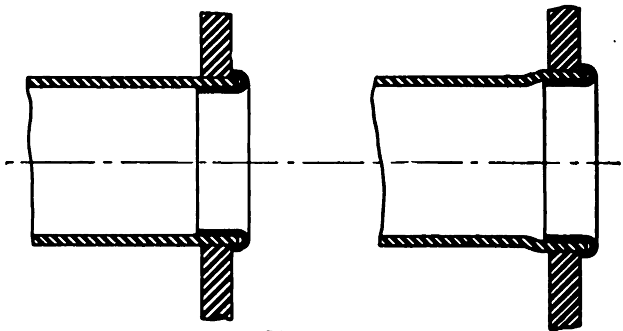


Fig. 114.

means the durability of the joint is enhanced and the tube ends are protected from the direct impact of the fire. To avoid choking the draft the tube ends are sometimes swaged out, as shown at the right in Fig. 114, to a diameter such that the smoke area is not contracted.

60. Stay Tubes. — When the boiler pressure much exceeds 150 lbs. per sq. in. and the tubes are spaced more than 2 ins. apart in the clear, it is necessary to adopt some other means than expanding and beading to insure the holding power of the latter as stays. For this reason tubes are often upset at the ends, threaded and screwed through the tube sheets, a check nut frequently being employed upon the outside, Fig. 115. With both ends thus secured to place, it is necessary to have the pitch of the threads continuous in both sheets. With long tubes this is somewhat difficult to obtain. After inserting threaded tubes, they are sometimes expanded and beaded. In large marine boilers every third or fourth tube, horizontally and vertically, is often made a stay tube. Also when the presence of internal water legs requires the omission of certain areas of tubes, those left along the margin are furnished with nuts to enhance their holding power. Tube ends should be upset when threaded so as to leave the root area substantially larger than that of the shank.

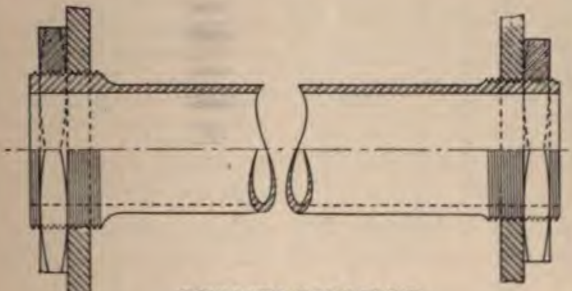

SCREWED STAY TUBE

Fig. 115.

The National Tube Company furnish upset and expanded boiler tubes of the five different patterns illustrated in Fig. 116, *a* to *e*. Only the first two forms with external and internal upsets are standardized. Tables XXXII and XXXIII on the following pages give the prescribed dimensions of these forms. Other sizes

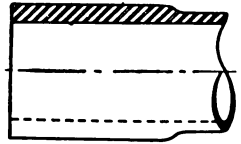


TABLE XXXII.

ADVISABLE EXTERNAL UPSETS for LAP-WELDED OR SEAMLESS TUBES.

National Tube Co.

Thickness of tube wall.		External diameter of tube. Ins.														
		1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾	4	4¼	4½	4¾	5
In.	Nearest B. W. G.	External diameter of upset. Ins.														
0.134	10	1.70	1.95	2.20	2.45	2.70	2.95	3.20	3.45	3.70	....	....	....	....	....	....
0.148	9	1.72	1.97	2.22	2.47	2.72	2.97	3.22	3.47	3.72	3.97	4.22	4.47	4.72	4.97	....
0.165	8	1.75	2.00	2.25	2.50	2.75	3.00	3.25	3.50	3.75	4.00	4.25	4.50	4.75	5.00	5.25
0.188	7	1.78	2.03	2.28	2.53	2.78	3.03	3.28	3.53	3.78	4.03	4.28	4.53	4.78	5.03	5.28
0.203	6	1.80	2.05	2.30	2.55	2.80	3.05	3.30	3.55	3.80	4.05	4.30	4.55	4.80	5.05	5.30
0.219	5	1.83	2.08	2.33	2.58	2.83	3.08	3.33	3.58	3.83	4.08	4.33	4.58	4.83	5.08	5.33
0.238	4	1.86	2.11	2.36	2.61	2.86	3.11	3.36	3.61	3.86	4.11	4.36	4.61	4.86	5.11	5.36
0.250	3	1.88	2.13	2.38	2.63	2.88	3.13	3.38	3.63	3.88	4.13	4.38	4.63	4.88	5.13	5.38
0.281	2	1.92	2.17	2.42	2.67	2.92	3.17	3.42	3.67	3.92	4.17	4.42	4.67	4.92	5.17	5.42
0.313	1	1.97	2.22	2.47	2.72	2.97	3.22	3.47	3.72	3.97	4.22	4.47	4.72	4.97	5.22	5.47
0.344	0	2.02	2.27	2.52	2.77	3.02	3.27	3.52	3.77	4.02	4.27	4.52	4.77	5.02	5.27	....
0.375	00	2.06	2.31	2.56	2.81	3.06	3.31	3.56	3.81	4.06	4.31	4.56	4.81	5.06	....	....
0.406	....	2.11	2.36	2.61	2.86	3.11	3.36	3.61	3.86	4.11	4.36	4.61	4.86	....	....	....
0.438	....	2.16	2.41	2.66	2.91	3.16	3.41	3.66	3.91	4.16	4.41	4.66	....	....	....	....

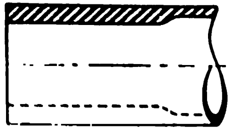


TABLE XXXIII.

ADVISABLE INTERNAL UPSETS
for
LAP-WELDED OR SEAMLESS TUBES.
National Tube Co.

Thickness of tube wall.		External diameter of tube. Ins.														
		1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾	4	4¼	4½	4¾	5
In.	Nearest B. W. G.	Internal diameter of upset. Ins.														
0.134	10	1.03	1.28	1.53	1.78	2.03	2.28	2.53	2.78	3.03	....	....	....	....	....	....
0.148	9	0.98	1.23	1.48	1.73	1.98	2.23	2.48	2.73	2.98	3.23	3.48	3.73	3.98	4.23	....
0.165	8	0.92	1.17	1.42	1.67	1.92	2.17	2.42	2.67	2.92	3.17	3.42	3.67	3.92	4.17	4.42
0.188	7	0.84	1.09	1.34	1.59	1.84	2.09	2.34	2.59	2.84	3.09	3.34	3.59	3.84	4.09	4.34
0.203	6	0.79	1.04	1.29	1.54	1.79	2.04	2.29	2.54	2.79	3.04	3.29	3.54	3.79	4.04	4.29
0.219	5	....	0.98	1.23	1.48	1.73	1.98	2.23	2.48	2.73	2.98	3.23	3.48	3.73	3.98	4.23
0.238	4	....	0.91	1.16	1.41	1.66	1.91	2.16	2.41	2.66	2.91	3.16	3.41	3.66	3.91	4.16
0.250	3	....	0.87	1.12	1.37	1.62	1.87	2.12	2.37	2.62	2.87	3.12	3.37	3.62	3.87	4.12
0.281	2	.	....	1.02	1.27	1.52	1.77	2.02	2.27	2.52	2.77	3.02	3.27	3.52	3.77	4.02
0.313	1	....	....	....	1.15	1.40	1.65	1.90	2.15	2.40	2.65	2.90	3.15	3.40	3.65	3.90
0.344	0	..	....	....	....	1.29	1.54	1.79	2.04	2.29	2.54	2.79	3.04	3.29	3.54	....
0.375	00	.	. .	....	....	....	1.44	1.69	1.94	2.19	2.44	2.69	2.94	3.19	....	....
0.406	....	..	...	....	.. .	....	....	1.58	1.83	2.08	2.33	2.58	2.83	....	....	....
0.438	....	.	....	...	. .		....	1.46	1.71	1.96	2.21	2.46	....	....	....	....

can be procured but their manufacture is attended with difficulty and increased expense. The tabulated dimensions are specified upon a basis of 2½ ins. length of upset.

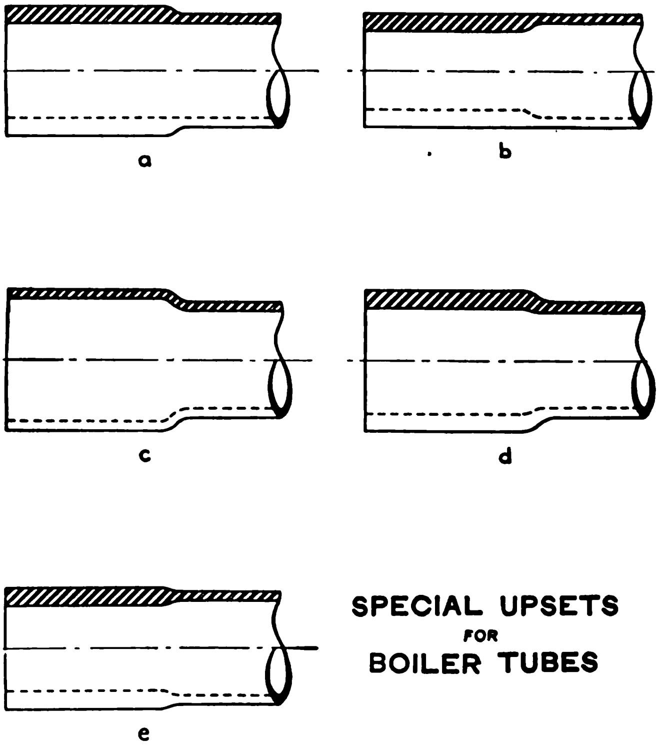


Fig. 116.

61. Strength of Boiler Tubes, Flues and Furnaces. — The theoretical discussion of tubes subjected to external pressure was given in Art. 19, page 63, Chap. II. Many practical rules and formulæ have been suggested for finding the external pressure which tubes of certain diameter and thickness will sustain. The best summary of such practice is found in the Rules of the U. S. Supervising Inspectors of Steam Vessels, Edition of Jan. 13, 1914. All tubes, flues and furnaces are divided into four classes as follows:

ss I. Plain lap-welded or seamless tubes up to and including 6 ins. external diameter.

ss II (*a*). Plain lap-welded or seamless flues over 6 ins. and not exceeding 18 ins. external diameter.

(*b*). Riveted flues over 6 ins. and not exceeding 18 ins. external diameter.

ss III. Riveted, seamless or lap-welded flues over 18 ins. and not exceeding 28 ins. in external diameter, made up in sections or courses.

ss IV. Special furnaces having walls reinforced by ribs, rings or corrugations.

Class I. — Tubes up to and including 6 ins. external diameter, of standard thickness, may be subjected to external pressures accordance with those given in the following table:

TABLE XXXIV.

MAXIMUM EXTERNAL WORKING PRESSURES

for use with

LAP-WELDED AND SEAMLESS BOILER TUBES.

From 2 to 6 ins. external diameter.

inal diam. ernal diam.	Standard thickness.	Maximum allowable pressure.	Nominal diam. External diam.	Standard thickness.	Maximum allowable pressure.
Ins.	In.	Lbs. per sq. in.	Ins.	In.	Lbs. per sq. in.
2	0.095	427	$3\frac{1}{2}$	0.120	308
$2\frac{1}{4}$	0.095	380	$3\frac{3}{4}$	0.120	282
$2\frac{1}{2}$	0.109	392	4	0.134	303
$2\frac{3}{4}$	0.109	356	$4\frac{1}{2}$	0.134	238
3	0.109	327	5	0.148	235
$3\frac{1}{4}$	0.120	332	6	0.165	199

If a thickness greater than the standard one is used, the ex- rnal working pressure may be calculated by the formula

$$t = \frac{(1386 + Fp)d}{86{,}670}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (124)$$

here t = thickness of tube wall, ins.,

F = factor of safety, to be taken as 5 for usual cases,

p = external working pressure, lbs. per sq. in.,

d = external diameter of tube, ins.

Class II (*a*). — Plain lap-welded or seamless flues, over 6 ins. and not exceeding 18 ins. external diameter and having no rivets except those at their end ring seams, may be subjected to the working pressures given in Table XXXV. It may be noted that the tabulation gives abstract thicknesses. There is a certain degree of standardization of tubes from 7 to 21 ins. external diameter, but the dimensions are not universally accepted. It is better, therefore, to stipulate abstract thicknessee and the corresponding pressures.

TABLE XXXV.

MAXIMUM EXTERNAL WORKING PRESSURES
for use with
LAP-WELDED AND SEAMLESS BOILER TUBES.
Over 6 ins. and not exceeding 18 ins. external diameter.

Nominal diam. Outside diam. of flue.	Thickness required for various pressures.						
	100 lbs. sq. in.	120 lbs. sq. in.	140 lbs. sq. in.	160 lbs. sq. in.	180 lbs. sq. in.	200 lbs. sq. in.	220 lbs. sq. in.
Ins.	In.	In.	In.	In.	In.	In.	In.
7	0.152	0.160	0.168	0.177	0.185	0.193	0.201
8	0.174	0.183	0.193	0.202	0.211	0.220	0.229
9	0.196	0.206	0.217	0.227	0.237	0.248	0.258
10	0.218	0.229	0.241	0.252	0.264	0.275	0.287
11	0.239	0.252	0.265	0.277	0.290	0.303	0.316
12	0.261	0.275	0.289	0.303	0.317	0.330	0.344
13	0.283	0.298	0.313	0.328	0.343	0.358	0.373
14	0.301	0.320	0.337	0.353	0.369	0.385	0.402
15	0.323	0.343	0.361	0.378	0.396	0.413	0.430
16	0.344	0.366	0.385	0.404	0.422	0.440	0.459
17	0.366	0.389	0.409	0.429	0.448	0.468	0.488
18	0.387	0.412	0.433	0.454	0.475	0.496	0.516

The thicknesses in the above table are calculated from formula (124) with a factor of safety of 5, and are applicable to lengths greater than six diameters of the tube, to working pressures greater than 100 lbs. per sq. in. and to temperatures not over 650° F.

Class II (*b*). — Riveted flues over 6 ins. and not exceeding 18 ins. external diameter, constructed of iron or steel plate not less than 0.25 in. thick, and put together in sections not less than 24 ins. in length, are calculated for maximum allowable external pressure by the following formula:

$$p = \frac{8100\,t}{d}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (125)$$

where p = external working pressure, lbs. per sq. in.,
t = thickness of plate, ins.,
d = external diameter of flue, ins.

Class III. — Riveted, seamless or lap-welded flues over 18 ins. and not exceeding 28 ins. external diameter, riveted together in sections not less than 24 ins. nor more than $3\frac{1}{2}$ times the flue diameter in length, and subjected to external pressure only, are calculated for maximum allowable external pressure by the following formula:

$$p = \frac{966\,t - 53\,l}{d}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (126)$$

where p = external working pressure, lbs. per sq. in.,
t = thickness of wall in sixteenths of an inch,
l = length of flue sections, ins., to be > 24 and $< 3\frac{1}{2}\,d$,
d = external diameter of flue, ins.

Class IV. — The strength of special flues or furnace tubes having walls reinforced by ribs, rings or corrugations has been the subject of much controversy and experiment. While it is an easy matter to deduce the theoretical stresses in cylinders subjected to external pressure, such calculations are of but little practical value. Any departure from true circularity in the cylinder will bring into play forces far in excess of those originally calculated. Hence all useful formulæ for this class of pressure vessels are in a sense empirical, depending for their value upon experimental coefficients.

The manufacture of furnace tubes is carried on in somewhat the following manner. A sheet of steel plate, sufficiently low in carbon to weld readily, is scarfed along two opposite sides and rolled by ordinary plate rolls to a true circle. Powerful flame jets of water gas are directed accurately between the overlapping scarfs, bringing them to a welding temperature. The joint is then carefully welded by machine throughout its entire length. The plain furnace tube thus formed, after being heated uniformly in an oven, is introduced between corrugating rolls. The latter operation gives the tube its final form and at the same time reduces its length a considerable amount in consequence of the corrugations. To secure different thicknesses of metal the plate used, as well as the distance between the corrugating rolls, is varied.

The earliest attempts to strengthen plain furnace tubes a
collapse took the form of angles or tees riveted to the exte
the tube in the form of rings. All such arrangements massed
volumes of metal close to the fire and the life of the exposed
heads was naturally of short duration. Most of these forn
now obsolete, having been superseded by the modern corru
furnace. The Rules of the U. S. Supervising Inspectors of
Vessels recognize three of these older types:

(*a*) Plain circular flues riveted together in sections.

(*b*) Plain circular flues flanged at the ends and rivet
Adamson rings.

(*c*) Circular flues made up of corrugated rings rivete
gether in sections.

The material specified for use in such furnace tubes must
a tensile strength lying between 58,000 and 67,000 lbs. per
and an ultimate elongation of 20 per cent in 8 ins.
otherwise stipulated the plain portions at the ends of tubes
kind must not exceed 9 ins. in length.

(*a*) *Riveted Tubes.* — While there is some enhanceme
strength from the double thickness of metal occasioned by m
up flues in sections, the advantage is largely offset by the
ability of overheating and consequent deterioration. Fo
pressures and moderate service the proportions for such
are calculated by formula (126) given on page 203. The thic
of wall in this formula must not be less than $\frac{5}{16}$ in., the
of sections not less than 18 ins. and the diameter of the fl
more than 42 ins. If any of the above conditions are exc
the wall of the flue must be stayed as a flat surface.

(*b*) *Adamson Rings.* — The Adamson ring, Fig. 117(a),
prises the most satisfactory method of reinforcing furnace
by means of riveted joints. The bulk of the metal compri
the ring and rivet heads is well removed from contact with th
The lap edges can be beveled and given a preliminary c
against the ring. There is also some flexibility in the
flanges adjoining the ring. The following proportions are
lated by the U. S. Supervising Inspectors:

Length of section not less than 18 ins.

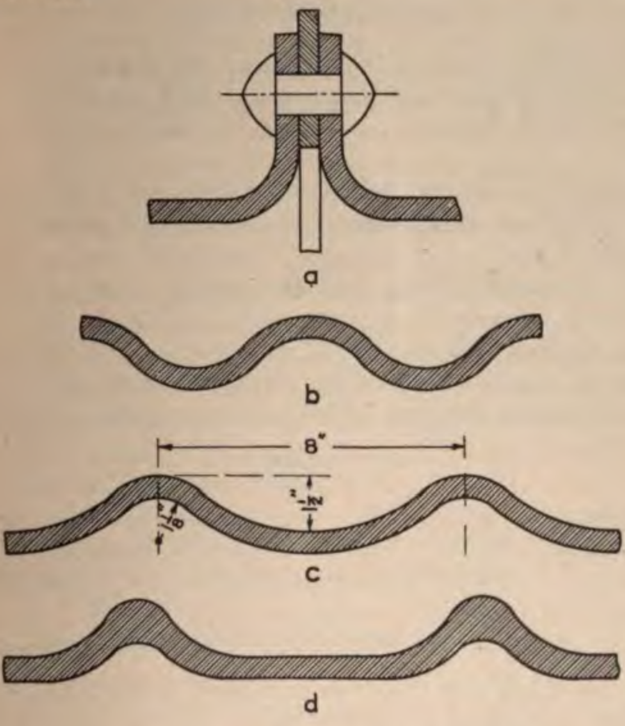
Plate not less than $\frac{5}{16}$ in. thick.

Radius of flange on fire side not less than three times the
thickness.

Depth of flange not less than radius on fire side plus three times the driven rivet diameter.

Thickness of flange as near that of plate as practicable.

Lap on flanges at least one and one-half times driven rivet diameter.



FURNACE CORRUGATIONS

Fig. 117.

Rivet diameter not less than plate thickness plus $\frac{1}{4}$ in.

Depth of ring not less than three times driven rivet diameter.

Thickness of ring not less than $\frac{1}{2}$ in.

When so designed, furnaces reinforced by Adamson rings may be subjected to pressures derived from the following formula:

$$p = \frac{1080\,t - 59\,l}{d}, \quad \ldots\ldots \quad (127)$$

where p = external working pressure, lbs. per sq. in.,
t = thickness of wall in sixteenths of an inch, not to be less than $\frac{5}{16}$ in.,
l = length of flue section, ins., not to be less than 18 ins.,
d = external diameter of flue, ins.

(*c*) *Corrugated Rings.* — Occasionally furnace tubes are made up of rings corrugated at the center and riveted together circumferentially. Such furnaces are subject to the following stipulations:

Length of sections not more than 18 ins.

Total depth of corrugations, including plate thickness, not less than $2\frac{1}{2}$ ins.

Ends accurately fitted into one another and substantially riveted together.

Plain parts at ends not over 12 ins. long.

When so constructed built-up corrugated flues may be calculated for external pressure by the following formula:

$$p = \frac{10{,}000\,t}{d}, \quad \dots\dots\dots \quad (128)$$

where p = external pressure, lbs. per sq. in.,
t = thickness of tube wall, ins., not to be less than $\frac{7}{16}$ in.,
d = mean diameter of cylindrical portion, ins.

The use of specially ribbed or corrugated furnaces has obtained exclusive acceptance in Scotch boilers. An attempt has been made, and with some success, to adapt such furnace tubes to the requirements of locomotive practice both in this country and in Europe. The exact form of the corrugation is an element of considerable importance and a vast number of types have been patented since the original effort of Mr. Samuel Fox in England. Of these the Fox, Morison, Purves, Brown, Leeds Bulb, Holmes, Farnley and Deighton have received more or less acceptance. The first three forms, however, are the ones most widely used, since they have been thoroughly tested and standardized by numerous Boards of Survey and Insurance. A specimen corrugation is given for each in Fig. 117(b), (c) and (d). It is to be noted that the metal is of uniform thickness in the Fox and Morison types while the Purves has ribs of increased depth. The formula used in calculating the external pressure on such furnaces is of the form

$$p = \frac{\text{Constant} \times t}{d}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (129)$$

where p = external working pressure, lbs. per sq. in.,
t = thickness of wall, ins., not to be less than $\frac{5}{16}$ in.,
d = mean diameter, ins., for Fox and Morison types and least outside diameter for Purves type.

The constant used in the numerator is the quantity derived from experiment and upon it the working pressure directly depends. For the Morison type the constant equals 15,600 and for the Fox and Purves, 14,000 respectively. The pitch of the corrugations is 6 ins. in the Fox, 8 ins. in the Morison and 9 ins. in the Purves type. There is an essential difference in the manufacture of these furnaces. Fox and Morison furnaces are made from flat plate corrugated after welding to a circular form. The proportions of the corrugations are always those shown in Fig. 117(b) and (c). The Purves furnace is ribbed while in the form of flat plate and is welded to a circular form thereafter.

The inside diameter of these members is usually calculated in connection with the necessary grate area. After having calculated the thickness, the outside diameter can be found by reference to Fig. 117. These furnaces are generally riveted to flanges bent out from the tube sheets, hence their ends are left plain. For marine use the reverse is the case in as much as the furnace ends are flanged and riveted to flat tube sheets. With plain ends the furnaces may be finished in either of three methods. First, both ends may be left slightly larger than the outer diameter of the corrugations to facilitate entrance through the tube sheet. A difference in diameter of $\frac{1}{4}$ in. between that of the corrugation and of the tube sheet opening is sufficient. Second, both ends may be finished to conform to the lesser diameter of the corrugations. This would require a somewhat smaller flanged opening in the rear tube sheet. Third, one end may be left large and the other small. In ordering furnaces the designer should take careful thought in regard to their renewal and provide for its easy accomplishment. The cylindrical portion at the ends should never be more than 9 ins. in length.

For marine use the combustion chamber end of the furnace is generally heavily flanged. To provide for entrance through the hole in the front tube sheet, the flanged end is made oval in outline with its lower lip bent up sufficiently to allow the furnace to be tipped into place.

Morison tubes, the type most widely used in this country, are obtainable in sizes varying in inside diameter from 28 ins. to 60 ins., and in thickness from $\frac{5}{16}$ in. to $\frac{3}{4}$ in. It is not good practice to specify tubes exceeding these limits.

Plenty of circulation space must be allowed around furnace tubes since they are rapid steam producers. When placed side by side the corrugations should be arranged to alternate so as to give a channel of uniform width between them.

62. Diameter of Boiler. — The ratio of steam space to water space is generally chosen more or less arbitrarily with reference to the type of boiler to be designed. A small water volume conduces to irregularity in steam pressure and necessitates careful attendance. In the reverse manner a large body of water steadies the operation of the boiler and serves as a reservoir of energy for sudden demands. A ratio of steam space to water space ranging from one-half to one-third is found to work well in practice. The total volume of an externally fired boiler is made up of three parts, namely, that occupied respectively by the tubes, steam and water. Having calculated the size and number of tubes, their total external volume should be set down as the first item. From the steam space per horse-power as specified in the problem at hand, the gross volume occupied by the steam may be computed. Lastly, with the stipulated ratio of steam space to water space in view, the total number of cubic feet of water in the boiler may be determined. The sum of the above three items taken in connection with the proposed length of tubes fixes the necessary diameter of the shell.

The length of boiler is generally established by the ratio of tube length to diameter as discussed under Art. 59, page 191. Very long boilers may be used when gas or oil is the fuel. They are, however, difficult to operate and are not as durable as those of less length. Table XXXVI gives a general idea of current practice in regard to boiler proportions.

From the requisite volume and specified length of the boiler the transverse area and corresponding diameter may be found. Boiler-

TABLE XXXVI.

USUAL PROPORTIONS OF BOILERS.

Horizontal Return Tubular Boilers.

International Engineering Co.

Diam. of shell.	Diam. of tubes.			Length of tubes.				
Ins.	Ins.			Ft.				
36	2½,	3	..	8,	10,	12	..	..
42	2½,	3	..	10,	12,	14	..	..
48	2½,	3	..	12,	14,	15,	16	..
54	3,	3½	..	13,	14,	15,	16,	18
60	3,	3½	..	14,	16,	18	..	..
66	3,	3½	..	16,	17,	18,	20	..
72	3,	3½,	4	16,	18,	20	..	..
78	3,	3½,	4	16,	17,	18,	20	..
84	3,	3½,	4	16,	18,	20	..	..
90	3,	3½,	4	16,	17,	18,	20	..
96	3,	3½,	4	16,	17,	18,	20	..

Vertical Fire-tube Boilers for Power Plant Use.

Diam. of tubes, ins.............................. 2, 2½

Length of tubes, ft.............................. 13, 14, 15, 16

Dry Back Scotch Boilers.

Continental Iron Works.

Diam. of shell.		Diam. of tubes.	Length of tubes.		Inside diam. of furnace.	Length of grate.	
			Short type.				
Ft.	Ins.	Ins.	Ft.	Ins.	Ins.	Ft.	Ins.
6	3	3	9	6	(1) 36	4	0
6	6	3½	11	6	(1) 36	5	0
7	0	3½	12	6	(1) 38	6	4
7	6	3½	12	6	(1) 45	6	8
8	0	3½	13	0	(1) 50	7	3
9	6	3½	12	6	(2) 38	6	4
10	6	3½	12	6	(2) 45	6	8
11	6	3½	13	0	(2) 50	7	3
			Long type.				
6	9	4	16	0	(1) 38	6	4
7	3	4	16	0	(1) 41	7	4
7	9	4	16	0	(1) 45	8	0
9	4	4	16	0	(2) 38	6	4
10	0	4	16	0	(2) 41	7	4
10	9	4	16	0	(2) 45	8	0

Locomotive Type Boilers Without Dome.

Hodge Boiler Works.

Diam. of waist.	Length of 3 in. tubes.		Dimensions of grate.			
			Width.		Length.	
Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.
36	7	0	2	6	4	0
42	8	0	3	0	4	2
48	10	6	3	6	4	6
54	11	0	4	0	5	0
60	13	0	4	6	5	0
66	15	0	5	0	5	6
72	15	0	5	6	6	0

making machinery is generally arranged to handle shells varying in diameter by multiples of half feet. Hence it is rarely wise to adopt boiler diameters other than those which are multiples of six inches. The next commercial diameter above that calculated should therefore be chosen. This sometimes secures a large increase in the volume of the steam space per horse-power.

The method outlined above for finding the boiler diameter is also followed for dry back Scotch boilers, except that the mean external volume of the furnace tube must be included with that of the fire tubes in determining the total volume. It is usually accurate enough to consider the various types of furnace tubes as plain cylinders, the diameter being that corresponding to the average depth of corrugations upon the outside of the tube.

In vertical boilers the procedure is somewhat different. The water leg surrounding the furnace must have a minimum net thickness of at least three inches in order to secure circulation. The circular grate must have a diameter sufficient to burn the requisite fuel economically. There must also be a narrow annular expansion space between the grate and internal furnace wall. The height of the furnace is generally made about three-fourths of its diameter to give room for combustion. Having calculated the necessary diameter of grate, the proposed boiler diameter may be determined by adding the room necessary for expansion and for the water leg accommodations. The total volume of a vertical boiler, Fig. 147, page 262, is made up of four parts, namely, the respective spaces occupied by the furnace, tubes, steam and water. Having assumed the dimensions suggested above in regard to height and diameter of furnace, its volume can be computed. To this should be added the aggregate external volume of the tubes. The length of tubes is usually fixed with reference to their diameter and set down as one of the original specifications. Having adopted a diameter and total height for the boiler the normal mean water level, based upon the specified ratio of steam to water, may be established and the actual volume of steam per horse-power determined. The maximum mean water level as fixed by the least permissible steam volume per horse-power should also be noted in order that the possible variation in the superheating surface may be obtained.

Vertical boilers are divided into three classes with respect to the form of shell: (*a*) straight shell boilers; (*b*) Manning or "ogee plate" boilers; (*c*) taper course boilers. The vertical straight shell boiler has a manhole in the middle course which permits inspection and repair. There is provided an annular space between the shell and tubes of sufficient width to enable a man to pass around the boiler inside. Constructively this type is superior to the other two, but the degree of superheat obtained is naturally very small.

The Manning boiler does not permit entrance for inspection and contains no stays in the upper head. In order to obtain a furnace of reasonable dimensions with the restricted steam and water space characteristic of this type, the lower course is increased in diameter by means of a reverse curve just above the fire box. The severe treatment to which boiler plate must be subjected in order to shape it to this complicated form is liable to cause structural weakness. There is also considerable endwise stress tending to straighten out the reverse curve. When built with the best of flanging machinery and careful workmanship, Manning boilers have served through long periods with safety and efficiency.

The taper course boiler is an attempt to gain the restricted steam space and consequent superheating efficiency of the Manning boiler without the use of the reverse curve. This type usually has a manhole and provides room for inspection in the lower part of the shell. A riveted joint in a tapering course is a difficult feature to calculate and still more so to construct. Hence the use of this type of boiler has been very limited.

The diameter of the cylindrical shell for locomotive-type boilers is calculated by the general method outlined above for horizontal return tubular boilers, except that an approximation must be made in regard to the steam and water volume around the furnace. As a rough estimate the latter volume may be assumed to constitute about one-fourth that of the entire boiler. If it is desired to obtain a final steam space of 0.8 cu. ft. per horse-power, the cylindrical portion of the shell should, therefore, be designed on the basis of three-fourths of the latter quantity or 0.6 cu. ft. per horse-power. After having laid out the shell, water legs and furnace, and established the mean water level for a specified ratio of steam to water, the exact volume of steam per horse-power should be computed. Water legs are generally given a net inside width of at least three inches and the pitch line of the mud ring seam is set from 25 per cent to 30 per cent of the boiler diameter below the exterior of the shell. With these proportions in view it is not difficult to calculate the final water volume and establish the exact ratio of steam to water. The contour of the furnace roof will be flat or curved, depending upon the type of staying to be used.

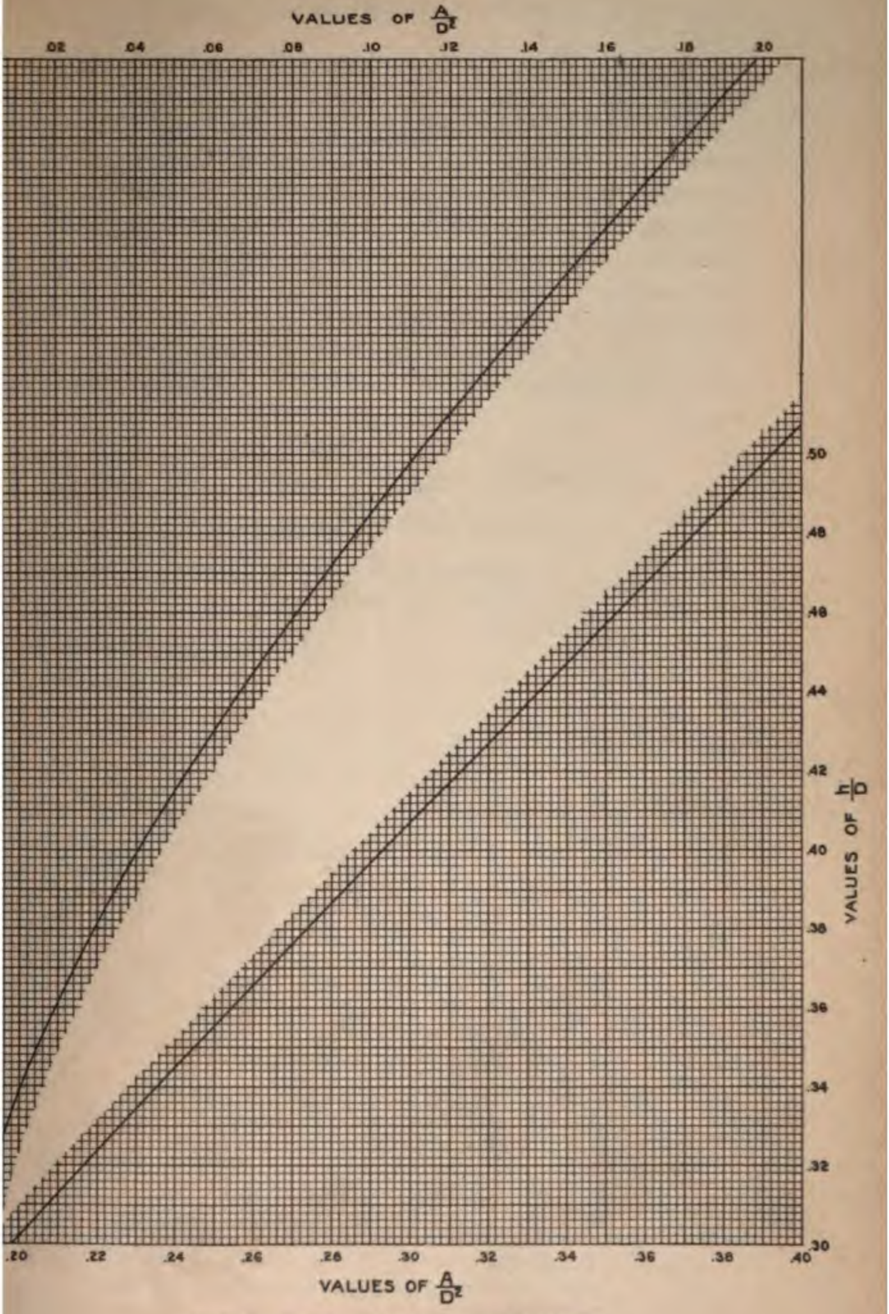
63. Steam Space. — The volume of steam space per horse-power depends upon the rapidity with which the steam is withdrawn, the possibility of using a dry pipe or steam separator and, lastly, the relative amount of disengaging surface at the normal water level. Restricted steam space conduces to wet steam and, if the boiler is rushed, to foaming. The usual volume per horse-power ranges from 0.6 to 0.8 cu. ft. per horse-power, and in rare instances the figure has been increased to 1 cu. ft. per horse-power. With the lesser of the above figures a dry pipe is generally used. Sometimes the requirement is specified that the volume of the steam space shall be adequate to supply the engine for 20 seconds, under the assumption that the water furnishes no steam during the period. Knowing the volume of the engine cylinder at cut-off and the revolutions per minute, the steam space may be calculated. Or, again, the steam space may be calculated from the specific volume of the steam and the number of pounds consumed per hour by the engine. The more rapid the withdrawal of steam and the smaller the disengaging surface, the greater should be the steam space.

64. Water Level. — Taking the diameter of the boiler as determined, the net volume may be divided into steam and water space in the specified ratio by reference to the accompanying curve, Fig. 118.

It is best in making this division to use the transverse area of the boiler rather than the volume. The curve is practically self-explanatory with the following nomenclature:

Let A = total area of steam segment,
D = diameter of shell or drum,
h = height of the segment to be occupied by steam.

Having found the gross transverse area of a proposed cylindrical boiler, the total external transverse tube area should be subtracted from it, leaving the net volume to be occupied by steam and water. Dividing this in the ratio of steam space to water space, the value of $\frac{A}{D^2}$ should be calculated and the percentage $\frac{h}{D}$ found by reference



AREAS OF CIRCULAR SEGMENTS

FIG. 118.

to the plot. This procedure locates the normal mean water level.

The distance between gage cocks varies somewhat with the diameter and horse-power of the boiler. Less than 3½ ins. of water above the top row of tubes is not safe and the lowest gage should be set at this uniform limit. When the disengaging surface is large, as in Scotch boilers, and small increments in the height of the segment correspond to large volumes of water, the gage cocks may be placed relatively close together. The reverse is true of those types of boilers having narrow confined steam spaces. Table XXXVII gives a general idea of current practice in this regard.

TABLE XXXVII.

WATER LEVEL REQUIREMENTS.

Horizontal Return Tubular Boilers.

Low water level 3½ ins. above surface of tubes.

Boiler diameters.	Distance between gage cocks.
Ins.	Ins.
36, 42, 48 ..	3
54, 60	4
66, 72, 78, 84	5

Vertical Fire Tube Boilers.

36 to 48 incl.	4
49 to 66 incl.	5
67 to 96 incl.	6
97 to 120 incl.	7

Dry Back Scotch Boilers.

Low water level 3½ ins. above surface of tubes for all diameters. Distance between gage cocks may be reduced to a minimum of 3 ins.

Locomotive Type Boilers.

Low water must be from 3 to 5 ins. above the upper surface of the crown sheet. The distance between gage cocks is usually 3 ins. for all diameters.

In order to avoid priming there must not only be sufficient steam space in the boiler but a reasonable amount of surface for the disengagement of the steam bubbles as well. In horizontal cylindrical boilers small changes in water level correspond to large variations in the amount of disengaging surface. The following table gives the average amount of disengaging surface per horse-power for a number of well designed boilers of various types.

TABLE XXXVIII.

DISENGAGING SURFACE PER HORSE-POWER.

Mean water level.

Type of boiler.	Disengaging surface, sq. ft. per H.P.
Horizontal return tubular	0.70 to 0.80
Dry back Scotch	0.60 to 0.70
Vertical straight shell	0.16 to 0.20
Vertical (Manning)	0.09 to 0.10
Locomotive-type	0.80 to 1.00
Sectional water tube	0.30 to 0.40

In order to make more evident the variation in disengaging surface corresponding to a given change in water level in horizontal boilers, the plot of Fig. 119 has been prepared. The abscissæ represent ratios of height of segment h to radius of shell R; and the ordinates similar ratios of the total width of disengaging surface w to the radius R.

Problem:

A 100 H.P. horizontal return tubular boiler, 60 ins. in diameter and 17 ft. long, carries a mean water level 17 ins. below the top of the shell. What is the disengaging surface per H.P.?

Solution: —

Value of $\frac{h}{R} = \frac{17}{30}$

$= 0.567.$

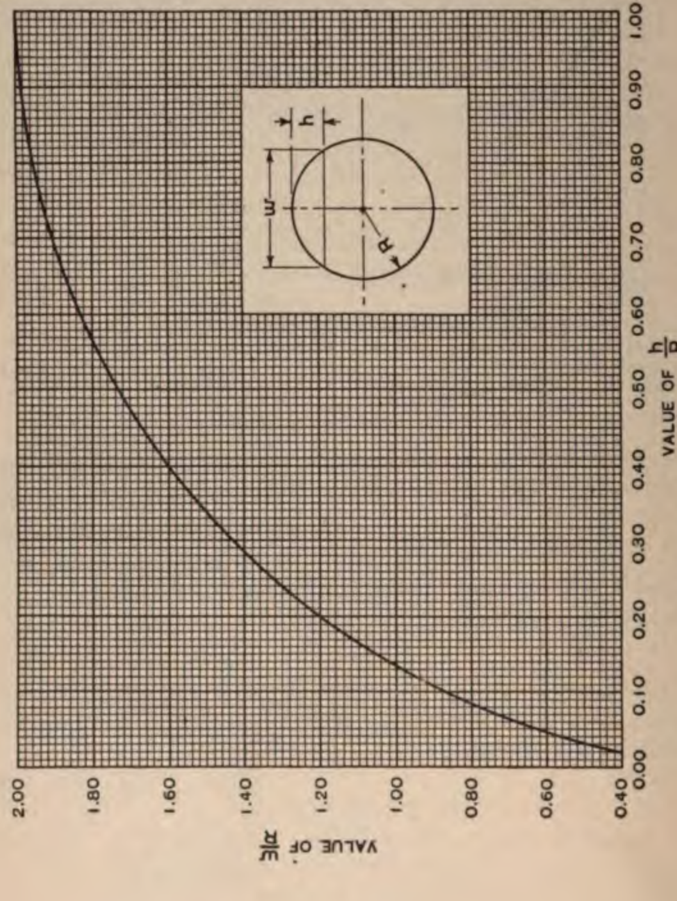
From Fig. 119, corresponding $\frac{w}{R} = 1.80$

or $w = 54.0$ ins.

Total disengaging area $= \frac{54}{12} \times 17$

$= 76.5$ sq. ft.

or 0.765 sq. ft. per H.P.



DISENGAGING SURFACE
IN
HORIZONTAL CYLINDRICAL SHELLS

Fig. 119.

The above requirements for disengaging surface have of necessity to be greatly reduced in the case of very large boiler horse-power units. As a limit one-half the above specifications are sometimes found in boilers of 800 and 1000 horse-power.

Having determined the position of the mean water line, the upper and lower gage cocks and the tops of the tubes should be located. It is frequently difficult to secure room enough for the requisite number of tubes, especially when there is a manhole in the front tube sheet. The water level in the boiler is often arbitrarily raised for this purpose. Taking the minimum volume required for the steam space of the boiler, the highest possible position of the tubes should next be calculated. Preparatory to laying out the tube sheet a drawing should be made of the front head of the boiler with the normal and maximum tube levels indicated upon it. The placing of the water level in vertical cylindrical boilers is an easy matter, merely requiring the subdivision of the total boiler volume in the ratio specified for steam and water.

65. Shell Thickness. — Taking into account the specified constants for boiler plate, a factor of safety of at least five and the efficiency of the proposed joint, the shell thickness may be read from the plot of Fig. 11, page 57. Boiler plate is readily procurable in thicknesses varying by $\frac{1}{16}$ in. In rare cases the thickness of the shell may be taken to the nearest thirty-second of an inch, but such plate has to be specially ordered. The next commercial thickness above the decimal value calculated should usually be selected. For the shells of externally fired boilers $\frac{9}{16}$ in. is generally considered the maximum allowable thickness, due to the liability of thicker plates to overheat and burn. Shells $\frac{9}{16}$ in. and $\frac{5}{8}$ in. thick, when necessary, present special difficulty at the ring seam. The mass of metal at this point is very liable to be injured by overheating. To obviate this various designers have cut away part of the plate thickness locally, as shown in Fig. 120. The tension on the ring seam being small in braced cylindrical shells, this expedient is a safe one. In the same manner a reduction of thickness is frequently made at the seam where furnace tubes

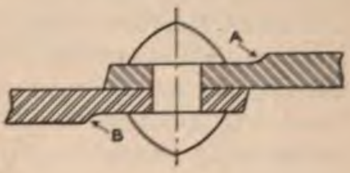
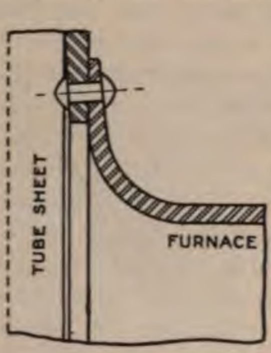


FIG. 120.

are joined to the tube sheets of Scotch boilers. Fig. 121 shows the method by which this is accomplished. The metal in this seam is especially liable to deterioration if left full thickness, consequently the scarfing of each plate is a desirable expedient. Abrupt changes of thickness must be carefully avoided however. Before adopting such an arrangement accurate calculations to determine the least factor of safety should be made, assuming no help from the braces, tubes or stay rods. The effect of corrosion should also be included in such an investigation, since ring seams are liable to this reduction in strength.



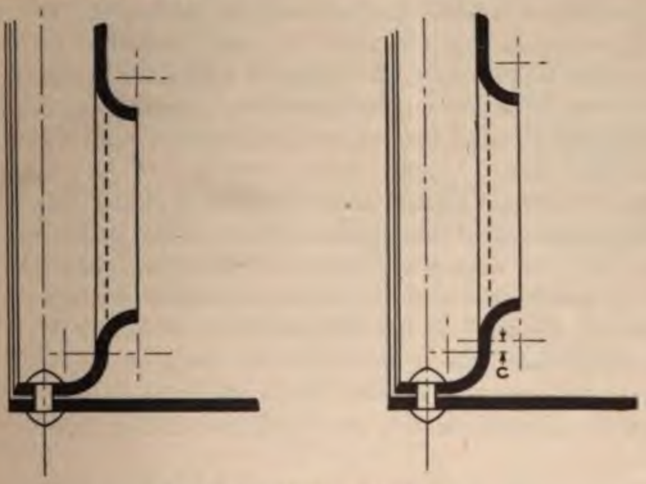
FURNACE SEAM
Fig. 121.

66. Tube Sheet. — Having determined the maximum and normal tube levels for horizontal cylindrical boilers, the detailed arrangement of the tube sheet should next be worked out. It should be kept in mind that the tubes are useful not alone in providing heating surface and in conveying away the smoke, but also as stay rods, the expanding and beading at their ends making them very effective for this purpose. Too much unstayed plate between the tubes and shell is to be avoided, since its presence would necessitate isolated local stays. Such arrangements are rarely to be tolerated especially in small boilers. From the reverse standpoint the tubes must be kept a reasonable distance away from the shell in order to insure some flexibility and provide for differing degrees of expansion. The water circulation also demands a clear channel outside of the tube space. Hence the tubes are generally arranged to fill as uniformly as possible the space below the maximum tube level bounded by the arc of a circle 3 ins. inside of the innermost course.

It is never wise to locate one or two isolated tubes low down in the sheet close to the shell. They possess but little value as heating surface, impede the circulation where the latter is most needed and displace a substantial amount of water which should be present to absorb the direct heat of the fire. It is a good rule to secure an uninterrupted segment of water not less than 6 ins. in depth over the furnace in all externally fired boilers.

Whenever man- and handholes occur care must be taken that the neighboring tubes do not encroach upon the room necessary for gaskets and flanges.

The thickness of the tube sheet is generally first assumed and later taken into account in designing the staying system. As a rule the tube sheet should be either $\frac{1}{16}$ in. or $\frac{1}{8}$ in. thicker than the shell for horizontal cylindrical boilers. Tube sheets $\frac{3}{8}$ in. thick are difficult to stay and those $\frac{5}{8}$ in. and over in thickness are too rigid, hence the thicknesses in general use are $\frac{7}{16}$ in., $\frac{1}{2}$ in. and $\frac{9}{16}$ in. The effect of the thickness adopted upon the staying calculation is explained in Art. 68.



MANHOLE OPENINGS
IN
FRONT HEAD
FIG. 122.

In modern boiler practice the flange at the margin of the tube sheet is rolled to shape with as little injury to the plate as possible. The tube sheet, heated to a bright red, is clamped in a flanging machine and its edge spun to the desired radius by internal and external rollers. Flanges thus formed may have an internal radius equal to the thickness of the plate as a minimum, but whenever possible two to three times the thickness should be allowed for the internal radius. Flanges so formed are a source of inherent rigidity to the adjacent plate.

Man- and handholes should be so located that their gaskets shall in all cases be provided with flat seats. Yokes or crabs should also be provided with adequate standing room upon the plate. Such openings should be placed as low down in the boiler head as the flanging operation above described will permit. When a flanged seat is provided for the manhole the reverse curve shown in Fig. 122, at the left, may be used as a minimum arrangement. It is better, however, when possible, to provide at least one-half inch of flat plate *c* between the two portions of the reverse curve as shown at the right. When considering the proximity of tubes to flanged openings care must be exercised to secure flat plate against which the operation of beading the tube may be performed. Sometimes the symmetry of the tube sheet is locally disturbed in order to throw a few tubes nearer the outline of a man- or handhole.

There are three general methods of tube arrangement for horizontal cylindrical boilers: (*a*) vertical rows; (*b*) diverging rows; (*c*) diagonal rows.

Fig. 123 shows a typical sheet of the first class. The vertical and horizontal spaces between tubes should never be less than ¾ in. in the clear, and when possible the horizontal ones should be made 1 in. A vertical circulation space at the center of the tube sheet is desirable. If used, its net width should be at least 2 ins. When more space than this is available care must be taken that the unstayed tube sheet in this locality is not liable to give way. In general the dimensions and ratios indicated in Fig. 123 should be given.

Any additional room upon the upper horizontal rows of tubes may be taken into good account to secure better circulation. Fig. 124 shows a typical tube sheet arrangement with diverging rows. The central rows are arranged as shown with vertical center lines parallel with that of the tube sheet. The top row is then spaced out arbitrarily, so as to bring the right hand tube close to the three inch limit. This may give spaces between the tubes of decimal value. An over all measurement, however, will enable the tubes to be placed without difficulty. Near the handhole the tube spacing in a horizontal direction is restricted to ¾ in. in the clear, and by stepping off the measurements *k k k*, a maximum number of tubes are accommodated. This arrangement places the maximum circulation space where the steam bubbles are the most numerous and conduces to dry steam and easy operation.

When it is very necessary to provide a maximum amount of

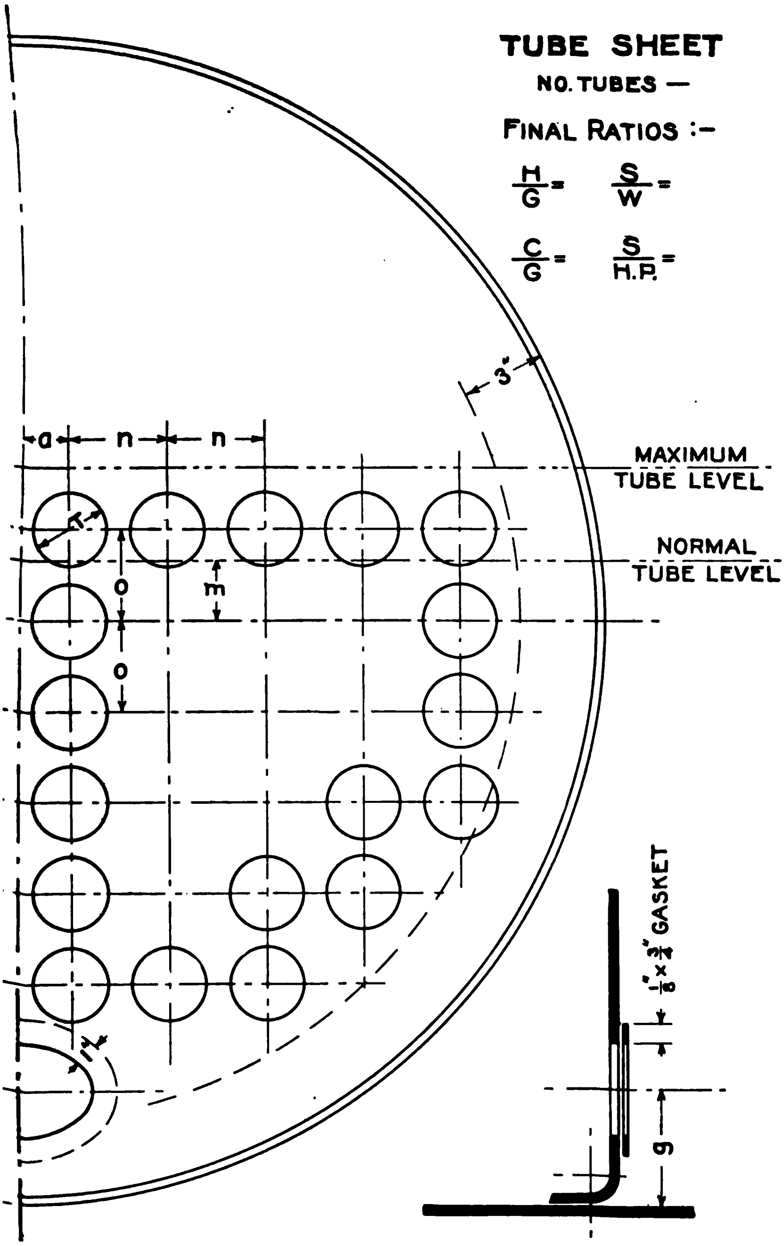

TUBE SHEET
NO. TUBES —
FINAL RATIOS :-
H/G =
S/W =
C/G =
S/H.P. =
3"
a
n
n
MAXIMUM TUBE LEVEL
NORMAL TUBE LEVEL
r
o
o
m
1/8" × 3/4" GASKET
g

Fig. 123.

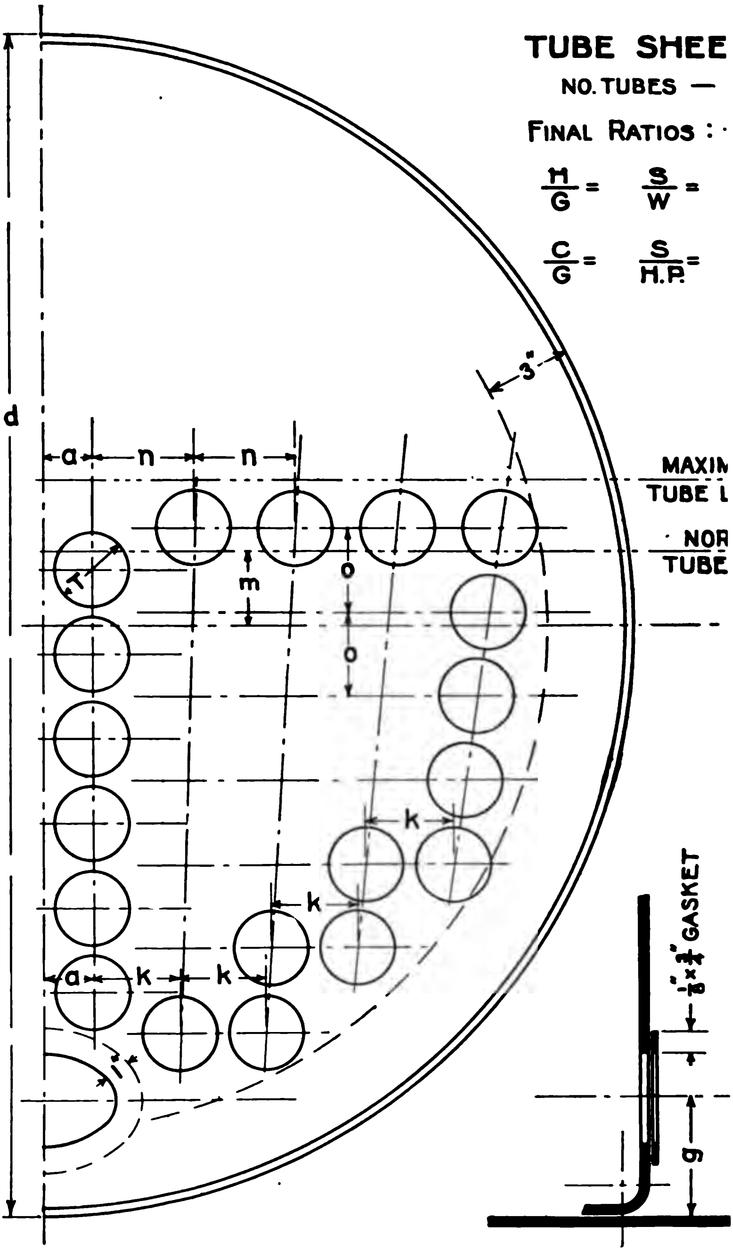

TUBE SHEE
NO. TUBES —
FINAL RATIOS :
H/G =
S/W =
C/G =
S/H.P. =
3"
MAXIM
TUBE L
NOR
TUBE
d
a
n
n
T
m
o
o
k
k
a
k
k
1"
1/8" × 3/4" GASKET
g

Fig. 124.

heating surface and the tube sheet area is of necessity restricted, the arrangement shown in Fig. 125 may be adopted. Where the boiler is in motion, such as in marine and locomotive practice, the steam bubbles will be detached from the tubes without special difficulty although the channels are as limited as shown. The distance between tubes in the clear should not be less than ¾ in. in any direction and the arrangement should be such that the sediment from the top of one tube will not lodge upon its neighbor. For stationary boilers diagonal tube sheets are rarely used. Broken or irregular arrangements of tubes are never permitted, since they tend toward complication and mistakes in the final layout.

In order to gain an idea of the possibility of a proposed arrangement, discs of paper or metal serving as templates to scale may first be laid down upon the drawing of the tube sheet outline. After such a trial an accurate location can be made. A complete vertical section from top to bottom of the tube sheet should be drawn, in addition to the elevation, for use in designing the staying later.

In dry back Scotch boilers, Fig. 126, the spacing of tubes is usually in vertical and horizontal rows. For marine use the tubes may be placed on diagonal lines. On account of the size of the boiler unit and the violence of the ebullition within, more space should be left between the tubes than in the case of horizontal return tubular boilers. At least 1¼ ins. horizontally and 1 in. vertically should be left in the clear between tubes. The furnace tube directly over the fire constitutes very efficient steam generating surface and therefore a central circulation space should always be provided in this type of boiler, its net width varying from five to ten inches with the diameter of the furnace tube. When two or more furnace tubes are used, it is well to have the corrugations arranged to come in alternation so that the circulation space between them may be of uniform width throughout. The distance in the clear between furnace tubes should not be less than 3 ins. and twice this amount is desirable when possible. The distance from the shell to the furnace tube may be determined the same as in the case of the manhole, Fig. 122, page 219.

In order that the tubes may end inside of the brick lining of the combustion chamber, the outside of the tube area should not approach nearer than 4½ ins. to the shell. The furnace tube

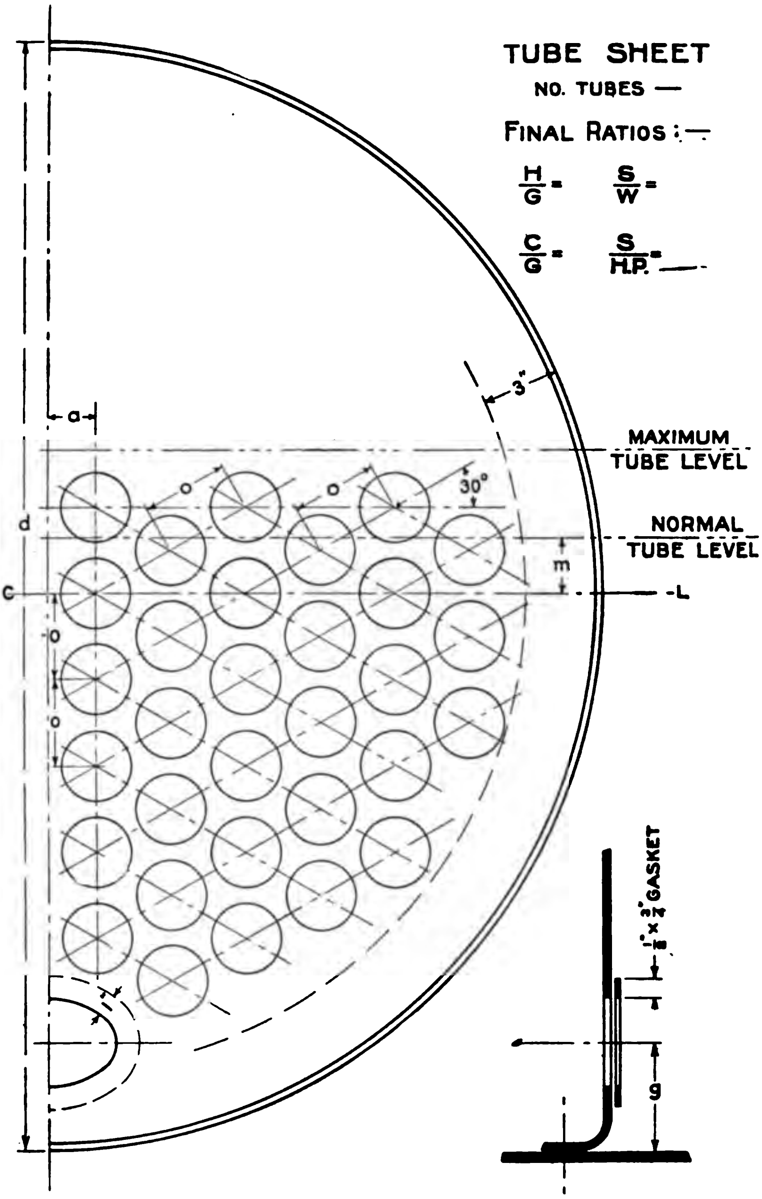

TUBE SHEET
NO. TUBES —
FINAL RATIOS :—
$\frac{H}{G}$ =
$\frac{S}{W}$ =
$\frac{C}{G}$ =
$\frac{S}{H.P.}$ = ——
3"
a
d
c
o
o
o
o
30°
m
MAXIMUM TUBE LEVEL
NORMAL TUBE LEVEL
L
1/8" × 3/4" GASKET
g

FIG. 125.

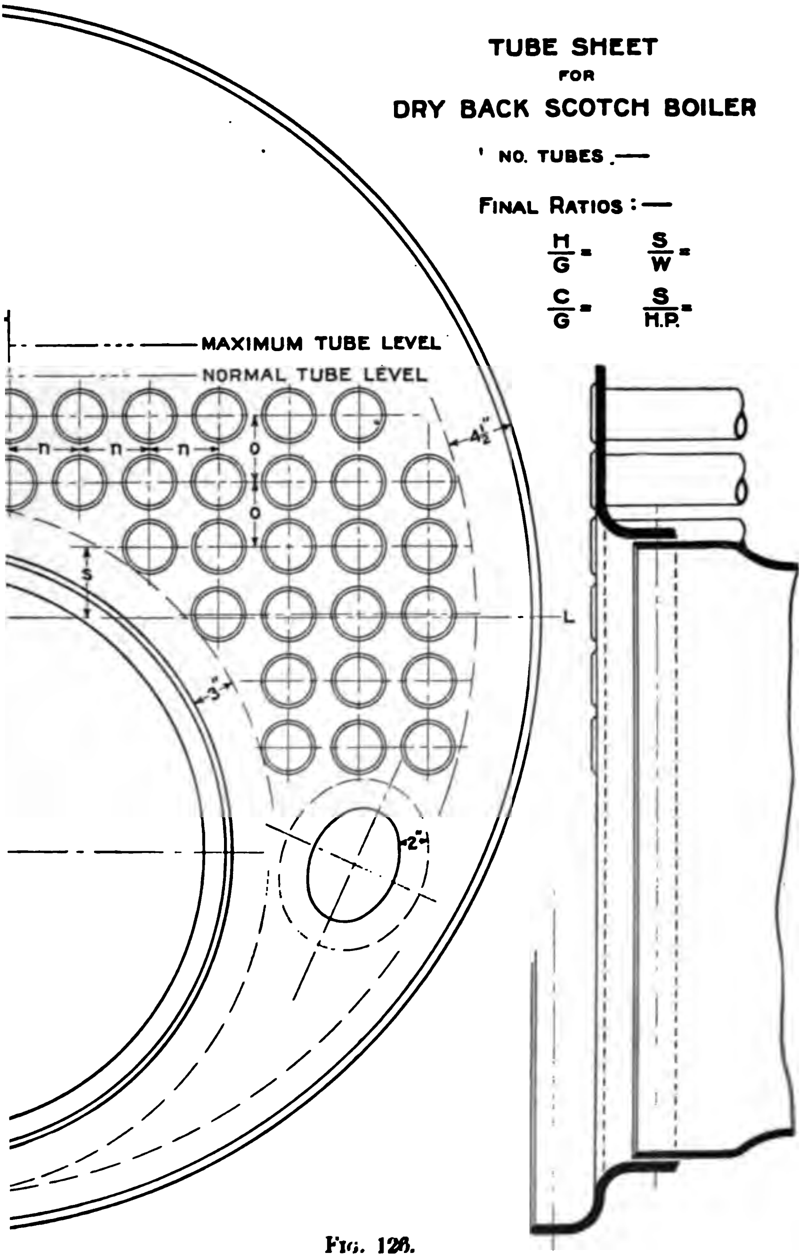

TUBE SHEET
FOR
DRY BACK SCOTCH BOILER
NO. TUBES —
FINAL RATIOS :—
H/G =
S/W =
C/G =
S/H.P. =
MAXIMUM TUBE LEVEL
NORMAL TUBE LEVEL
n
n
n
o
o
4½"
s
L
3"
2"

FIG. 126.

expands and contracts considerably, hence no fire tube should approach nearer than 3 ins. to the outside diameter of the corrugations. On account of the crowded steam space and large disengaging surface in this type of boiler the gage cocks are generally placed but 3 ins. apart, and 3½ ins. of water over the surface of the tubes is considered a sufficient amount for low water. Since the tube sheet above the tubes does not come under the direct influence of the fire except in a very limited degree at the rear head, much greater thicknesses of metal may be employed than in the case of externally fired boilers.

Two types of tube sheets are used in vertical boilers, depending upon their location with regard to the water level. Fig. 127 shows an immersed or sunken tube sheet at the right and one of the standard pattern at the left. With the immersed head there is much less difficulty in making the tubes tight, and at the same time the life of the tubes and upper tube sheet is greatly prolonged. The staying of the tapering section or "cone top" is so difficult, however, that this type of sheet is rarely attempted for boilers of more than 50 horse-power.

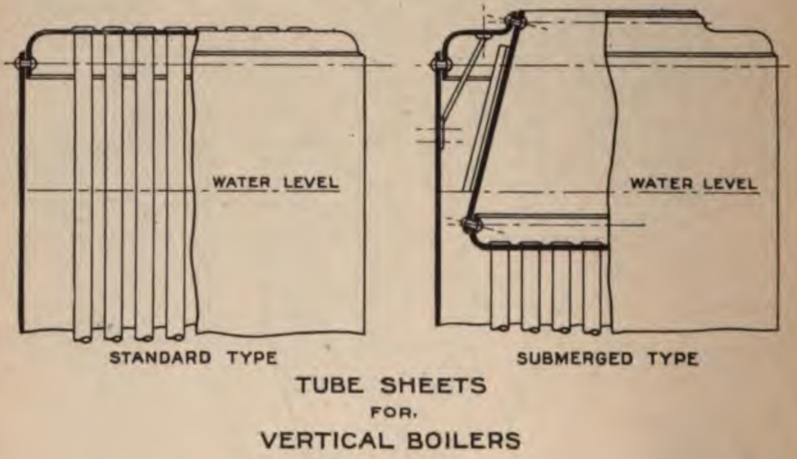


Fig. 127.

Standard tube sheets in vertical boilers usually have their tubes arranged in parallel rows to facilitate cleaning, Fig. 128. It is good practice to leave one inch in the clear around each tube with the centers arranged on lines at right angles to one another. In

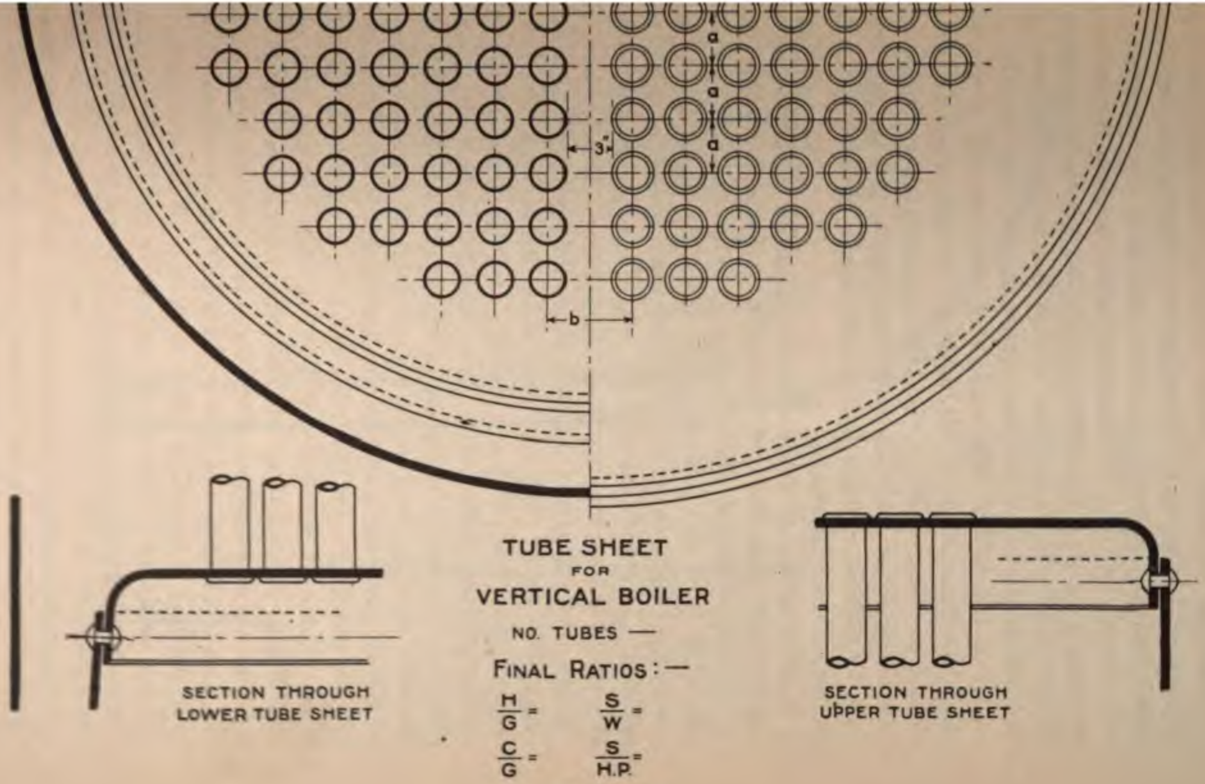

a
a
a
3"
b
TUBE SHEET
FOR
VERTICAL BOILER
NO. TUBES —
FINAL RATIOS: —
H/G =
S/W =
C/G =
S/H.P. =
SECTION THROUGH LOWER TUBE SHEET
SECTION THROUGH UPPER TUBE SHEET

Fig. 128.

the shell just above the furnace, numerous small handholes should be located through which the sediment which collects between the tubes may be removed. It is a good plan also to locate two avenues or "streets" through the tubes, at right angles to one another, communicating directly with four handholes in the shell. Such avenues should be 3 ins. wide in the clear. They are useful in cleaning and repair jobs, and permit the feed pipe to enter to the center of the boiler. The margin of plate outside of the tubes should be from 10 to 14 ins. wide for straight shell boilers and from 3 to 4 ins. wide for the Manning and taper course types. The boundary of the tube area should leave this margin as uniform as possible in order that a good degree of flexibility may be secured in the tube sheet.

In order to obtain the maximum amount of heating surface for a given boiler diameter, the tube sheets of locomotive-type boilers are usually arranged with diagonal rows as shown in Fig. 125, page 224. It is also possible in this manner to approximate more closely the tube sheet outline than when vertical and horizontal rows are used. In order to avoid stays within the tube area it is necessary that the latter shall approach the outline of the tube sheet with considerable regularity. Fig. 129 shows the design of a locomotive boiler tube sheet. The curvature of the furnace roof is generally assumed after the tubes are placed. It is thus made to fit their outline with fair uniformity. Since the furnace roof and adjacent tube sheet are heated by the fire to approximately the same degree, the tubes may approach the tube sheet flange to the closest possible proximity, provided a suitable amount of flat plate be secured for beading purposes. On the other hand the proximity of the tubes to the outer shell of the boiler must be limited by a margin of at least 3 ins. The hot tubes will expand to a far greater degree than the cool shell and sufficient flexibility must be secured in the smokebox tube sheet for this purpose. As in previous horizontal boilers, when possible, a central circulation space from 2 ins. to 2½ ins. in width is desirable. To facilitate cleaning, a handhole of standard dimensions must be located low down in the smoke box tube sheet. The proximity of the tubes to the handhole must not encroach upon the room necessary for the gasket. A minimum distance in the clear of ¾ in. must be secured between the tubes both in a vertical and diagonal direction in order to provide the necessary room for circulation.

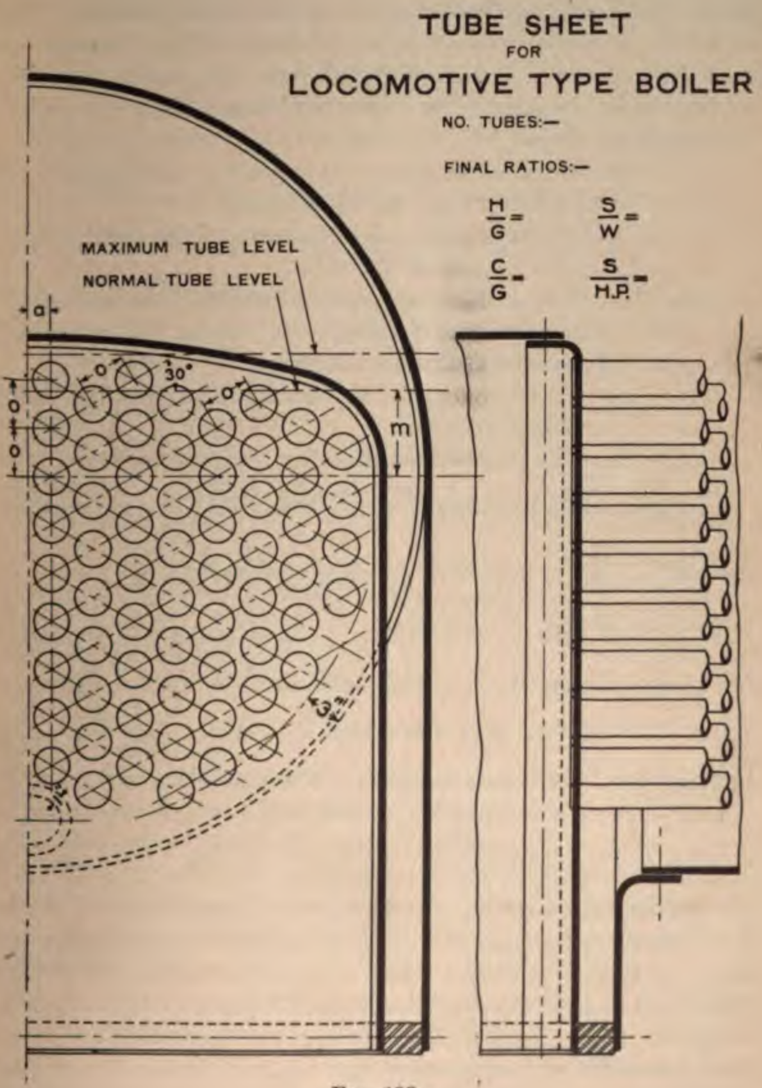

TUBE SHEET
FOR
LOCOMOTIVE TYPE BOILER
NO. TUBES:—
FINAL RATIOS:—
$\frac{H}{G}=$
$\frac{S}{W}=$
$\frac{C}{G}=$
$\frac{S}{H.P.}=$
MAXIMUM TUBE LEVEL
NORMAL TUBE LEVEL
a
o
o
o
o
30°
m

Fig. 129.

67. Final Ratios. — The relations between the heating surface, smoke and grate areas, and the steam and water volumes are generally noted in the specifications for a given design. They should be calculated after a final arrangement of the tubes has been made and accepted, so that they will represent actual conditions in the finished design. The following designations are used in expressing the ratios:

H = the total heating surface in contact with the fire,
G = area of the grate as finally adopted,
C = calorific or smoke area consisting of the total internal transverse area of the tubes,
S = the volume of steam included between the shell and a horizontal line through the position of the central gage cock as finally determined,
W = net water volume in the boiler below the line of the central gage cock,
H.P. = rated horse-power of boiler.

$\frac{\text{H}}{\text{G}}$ ranges from 35 to 45 in fire tube boilers, 37 being a good working value. $\frac{\text{S}}{\text{W}}$, as previously cited, should lie between $\frac{1}{2}$ and $\frac{1}{3}$ for most types of cylindrical boilers. $\frac{\text{C}}{\text{G}}$ varies from $\frac{1}{6}$ to $\frac{1}{8}$, the former for a short direct circuit of the gases and the latter for furnaces where more efficient heat absorption is sought. $\frac{\text{S}}{\text{H.P.}}$ may fall as low as 0.6 but 0.7 is a better limit where possible.

In addition to the above, various other ratios are frequently specified. The heating surface per horse-power or $\frac{\text{H}}{\text{H.P.}}$ is a relation upon which the capacity of the boiler is often based. The relative value of heating surface in the different types of steam generators has been the subject of much discussion. While externally fired boilers are generally provided with from 10 to 12 sq. ft. of heating surface per horse-power the latter figures are rarely reached with internally fired boilers. Without doubt the heating surface of furnace tubes and combustion chambers is from three to four times as efficient as that of smoke tubes and shells. Wherefore, Scotch boilers rarely exhibit more than 8 sq. ft. of heating surface per horse-power. With proper design and handling the latter figure is sufficient for a satisfactory boiler performance. In computing heating surface, the actual surface against which the fire strikes should be reckoned and not the surface in contact with the water. In the case of fire tubes this makes a considerable difference.

The ratio of the uptake area to that of the grate is usually made $\frac{1}{8}$. With tight connections and tall stacks this figure can be greatly reduced.

Some boiler makers specify the ratio of horse-power to grate surface or $\frac{\text{H.P.}}{\text{G}}$. With an evaporation of 9 lbs. of water per pound of dry coal and combustion at the rate of 20 lbs. per sq. ft. of grate surface per hour, 1 sq. ft. of grate surface will develop about $5\frac{1}{4}$ H.P. This value is directly dependent upon the relation of the ratios given above.

Since

$$\text{H.P.} \times 34.5 = \text{total water}$$

$$\frac{\text{H.P.} \times 34.5}{9} = \text{total coal}$$

or

$$\frac{\text{H.P.} \times 34.5}{9 \times 20} = \text{grate surface.}$$

Then

$$\begin{aligned}\frac{\text{H.P.}}{\text{G}} &= \frac{180}{34.5} \\ &= 5.22.\end{aligned}$$

To the above ratios should be added that of the disengaging surface per horse-power as explained under Fig. 119, page 216.

68. Staying. — One of the most important items in the design of any pressure vessel is the provision made for the rigidity of flat surfaces. In horizontal cylindrical boilers the upper portion of the tube sheet constitutes a surface of this class and great care must be exercised to insure its strength and durability.

It is assumed that all the upper portion of the tube sheet included between a horizontal line one-fourth the tube diameter above the center of the top row of tubes and a circle drawn where the tube sheet flange begins is flexible plate and provision must be made for its support. The determination of this area is shown in Fig. 130. When the stability of other portions of the tube sheet is in question the same method may be used to determine its boundaries. The beading of tubes is a source of considerable strength in their staying power, but it is an open question as to how far from their centers such rigidity extends. The above assumption is considered a conservative estimate of their holding power. The method just outlined is believed to be applicable to all sheet flanges having an internal radius of not over 3 ins.

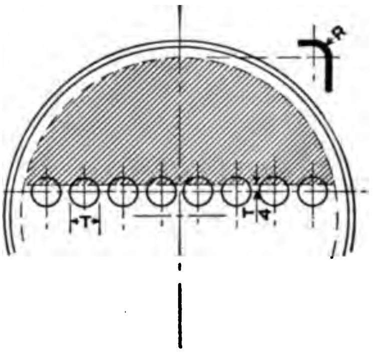


FIG. 130.

Two methods of staying are in practical use:

(a) *Through Rods.*
(b) *Diagonal Braces.*

Most of the calculations for the latter form are covered by the discussion of the former type, hence through rods will be taken up first.

(a) *Through Rods.* — Three limits will be assumed in placing stay rivets so that a reasonable degree of flexibility in the head shall be secured.

(1) *No rivet shall be less than 3 ins. from top of tubes or shell.*
(2) *No rivet shall be less than $2\frac{1}{2}$ ins. from a neighboring rivet.*
(3) *No through stay rod shall be less than 6 ins. from the shell.*

These limits should be regarded as prime essentials throughout the design.

The procedure in designing through stays should be carried out according to the following order:

I. Maximum spacing as regards plate.
II. Maximum spacing as regards rivets.
III. Arrangement of channels or angles.
IV. Arrangement of rods.
V. Final test for fibre stress in channels or angles.

I. Plate Spacing.— In expression (69), page 78, is given Grashof's formula for the maximum pitching of rivets or stay bolts spaced on squares when considered from the standpoint of the stress in the plate. It is necessary in stayed surfaces of this character not only to prevent the rupture of the plate but to keep the deflection small as well, hence a factor of safety of not less than ten should be employed. Taking the modulus of rupture of the flat steel plate as equal to its tensile strength the working fibre stress permissible for use in Grashof's formula would be 5500 lbs. per sq. in. Proceeding upon this basis the following table of maximum pitchings for various plate thicknesses and pressures has been computed:

TABLE XXXIX.

MAXIMUM SPACING

Of Stayed Points on Squares.—Ins.

Derived from Grashof's Formula.

Plate thickness. Ins.	Working Pressure. Lbs. per sq. in.								
	50	75	100	125	150	175	200	225	250
$\frac{1}{4}$	5.56	4.54	3.93	3.52	3.21	2.97	2.78	2.62	2.49
$\frac{5}{16}$	6.95	5.68	4.92	4.40	4.01	3.72	3.48	3.28	3.11
$\frac{3}{8}$	8.34	6.81	5.90	5.28	4.82	4.46	4.17	3.93	3.73
$\frac{7}{16}$	9.73	7.95	6.88	6.16	5.62	5.20	4.87	4.59	4.35
$\frac{1}{2}$	11.13	9.08	7.87	7.04	6.42	5.95	5.56	5.24	4.98
$\frac{9}{16}$	12.51	10.22	8.85	7.91	7.22	6.69	6.26	5.90	5.60
$\frac{5}{8}$	13.91	11.36	9.83	8.80	8.03	7.43	6.95	6.55	6.22
$\frac{11}{16}$	15.30	12.49	10.82	9.67	8.83	8.17	7.65	7.21	6.84
$\frac{3}{4}$	16.69	13.63	11.80	10.56	9.63	8.92	8.34	7.87	7.46
$\frac{13}{16}$	18.08	14.76	12.78	11.43	10.44	9.66	9.04	8.52	8.09
$\frac{7}{8}$	19.47	15.90	13.77	12.31	11.24	10.41	9.73	9.18	8.71
$\frac{15}{16}$	20.86	17.03	14.75	13.19	12.04	11.15	10.43	9.83	9.33
1	22.25	18.17	15.73	14.07	12.85	11.89	11.13	10.49	9.95

Having recorded the maximum permissible pitching for the thickness of tube sheet and steam pressure in the case at hand the second feature may be taken up.

II. Rivet Strength.—From the nature of the stresses in the head of a boiler and the severe effects of expansion and contraction it is evident that stay rivet material must be of the softest and toughest kind. Either wrought iron or very soft steel is used for this purpose. Taking the tensile strength from 45,000 to 55,000 lbs. per sq. in. and a factor of safety of 8, a working fibre stress f_t = 6000 lbs. per sq. in. may be used.

Let

b = side of the square which the rivet will support,

d = driven diameter of stay rivet,

p = working pressure in the boiler,

f_t = working tensile strength of rivet material, then

$$b^2p = \left(\frac{\pi d^2}{4}\right)f_t$$

or

$$b = \sqrt{\left(\frac{\pi d^2}{4}\right)\frac{f_t}{p}}$$

$$= \frac{d}{2}\sqrt{\frac{\pi f_t}{p}}. \qquad (130)$$

With the above value for rivet strength Table XL has been calculated, giving the side of square and the allowable area of plate which may be alloted to each stay rivet with various shank diameters and boiler pressures.

Having chosen a tentative rivet diameter the corresponding quantities from the above table should be recorded before proceeding to the third step in the design.

III. Arrangement of Girders. — When using through stay rods, channels or angles are riveted more or less intimately to the tube sheets above the tubes. The stay rods with nuts upon their external ends support the girders which these channels constitute, and bind the heads of the boiler together. Check nuts half as thick as standard ones are used on the inside to preserve staunchness and prevent leakage around the rod ends. A thin copper gasket is generally calked under the external stay rod nut for the same purpose.

Fig. 131 illustrates the general procedure in staying the flat head of a horizontal cylindrical boiler with through rods. It is always desirable to have the distances A and B between successive horizontal rows of rivets as nearly equal as possible. This secures uniform deflections and fairly flat surfaces against which the stay rod nuts may rest. The distance from the upper row of rivets to the top of the flange curve must be made considerably larger than the spaces between rows in order to keep the stay rods back the required distance from the shell. The diameter of the boiler will determine roughly whether one, two or three girders shall be used in the upper part of the tube sheet. In horizontal cylindrical boilers from 48 to 72 ins. in diameter two girders are entirely adequate. In boilers of the Scotch type there is sometimes a necessity of using three girders.

Having made a tentative vertical division with $A = B$, Fig. 131, and a reasonable allowance extra at the top, a channel of suitable

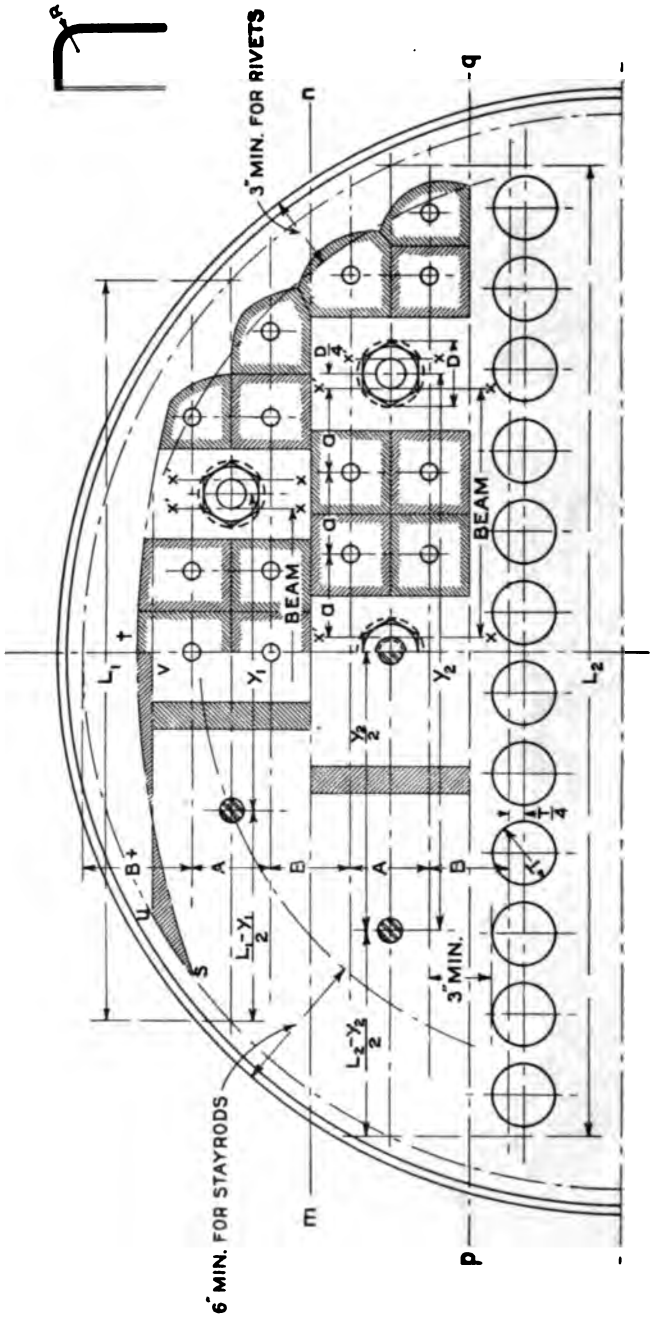
TABLE XL.

AREA AND SIDE OF SQUARE

That One Rivet will Support.

Working tension on rivet, 6000 lbs. per sq. in.

Pressure. Lbs. per sq. in.	Driven diameter of rivet. Ins.													
	5/8		11/16		3/4		13/16		7/8		15/16		1	
	Area.	Side of square.	Area.	Side of square.	Area.	Side of square.	Area.	Side of square.	Area.	Side of square.	Area.	Side of square.	Area.	Side of square.
	Sq. ins.	Ins.	Sq. ins.	Ins.	Sq. ins.	Ins.	Sq. ins.	Ins.	Sq. ins.	Ins.	Sq. ins.	Ins.	Sq. ins.	Ins.
50	36.81	6.07	44.54	6.67	53.02	7.28	62.22	7.89	72.16	8.50	82.83	9.10	94.23	9.71
75	24.54	4.96	29.70	5.45	35.34	5.95	41.48	6.44	48.10	6.94	55.22	7.43	62.82	7.93
100	18.41	4.29	22.27	4.72	26.51	5.15	31.11	5.58	36.08	6.01	41.42	6.44	47.12	6.86
125	14.72	3.84	17.82	4.22	21.20	4.61	24.89	4.99	28.87	5.37	33.13	5.76	37.70	6.14
150	12.27	3.50	14.85	3.85	17.67	4.21	20.74	4.55	24.05	4.91	27.62	5.26	31.42	5.61
175	10.52	3.24	12.73	3.57	15.15	3.89	17.78	4.22	20.62	4.54	23.67	4.87	26.93	5.19
200	9.20	3.03	11.14	3.34	13.25	3.64	15.55	3.95	18.04	4.25	20.71	4.55	23.56	4.85
225	8.18	2.86	9.90	3.15	11.79	3.43	13.82	3.72	16.03	4.01	18.41	4.29	20.94	4.58
250	7.36	2.71	8.91	2.99	10.60	3.26	12.45	3.53	14.43	3.80	16.57	4.07	18.85	4.34



TUBE SHEET STAYING
GIRDERS AND THROUGH RODS
Fig. 131.

size should be selected from the following table. Next the possibility of driving between the channel flanges two rivets of the chosen diameter, when placed A inches apart, should be determined. Table XLI gives all the usual standard rollings of channels suitable for this purpose. It may be assumed that rivets can be driven satisfactorily so long as their heads rest on the flat web inside of the channel.

When it is very desirable to keep the rivets as far apart as possible the rivet heads, taken as $2d$ in diameter, may be allowed to encroach a little upon the fillets, the minimum distance from the flange being $\frac{1}{8}$ in. as shown in Fig. 132. In any event the rivets should be spread as far apart as possible so as to load the stiff flanges rather than the web. In spite of these precautions A is often found to be less than B. A slight inequality is permissible, but if too great, a channel may be built up more nearly to suit the requirements of the case from some of the angles listed in Tables XLII and XLIII. While it is best to place the angles touching each other so as to present a good seat for the check nuts, sometimes a much better rivet arrangement may be secured by spacing them a small distance apart. The distance between them should not exceed $\frac{3}{8}$ in. Tables XLII and XLIII give all the standard rollings of angles suitable for use in boiler and tank staying. Since corrosion is liable to remove $\frac{1}{16}$ in. from the surface of such internal members of a boiler, it is not wise to use angle or channel thicknesses less than $\frac{3}{8}$ in. Angles are rolled without sloping sides, thus securing a good seat for the check nuts and obviating the use of special washers. Care must be taken that the rivet heads do not encroach too far upon the fillets as in the case of channels. Various special rollings of angles and channels may be found by consulting the handbooks of steel companies. Such sections are not included in the [illegible]

IV. *Arrangement of Rods.* In order that a man may enter the boiler for inspection it is necessary that through stays be [illegible] at least 16 ins. apart on centers under the manhole opening. This requirement forbids the use of through rods in [illegible] of [illegible] diameter unless braced apart.

More than two stay rods are [illegible] except in boilers of very large [illegible] the general usage to [illegible] central rod as far below [illegible] the case, does not [illegible]

TABLE XLI.

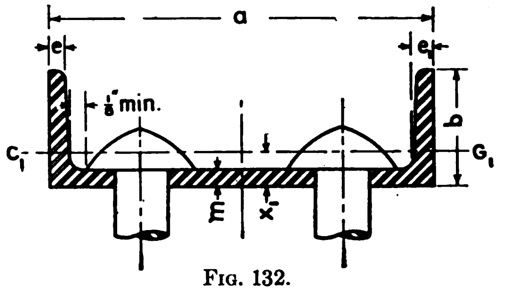


FIG. 132.

PROPERTIES OF STANDARD STRUCTURAL AND SHIP BUILDING CHANNELS

for use in Boiler Staying.

a Width of channel.	b Height of flange.	e Thickness of flange at top.	e_1 Thickness of flange at bottom.	Radius of fillet.	m Thickness of web.	x_1 Distance to C. G.	$I_{c.g.}$ Inch units.	$\frac{I_{c.g.}}{b-x_1}$ Inch units.	Weight per foot.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.			Lbs.
6	1.920	0.200	0.487	0.30	0.200	0.52	0.70	0.50	8.0
6	2.038	0.200	0.487	0.30	0.318	0.50	0.88	0.57	10.5
6	2.160	0.200	0.487	0.30	0.440	0.52	1.10	0.65	13.0
6	2.283	0.200	0.487	0.30	0.563	0.55	1.30	0.74	15.5
6	2.563	0.303	0.488	0.34	0.313	0.74	2.10	1.10	12.5
6	2.813	0.280	0.500	0.50	0.313	0.81	2.60	1.30	13.0
6	3.063	0.280	0.500	0.50	0.563	0.80	3.50	1.60	18.1
6	3.500	0.340	0.410	0.25	0.350	1.08	5.20	2.10	15.0
6	3.560	0.460	0.530	0.25	0.410	1.18	6.80	2.90	19.0
6	3.685	0.460	0.530	0.25	0.535	1.16	7.80	3.10	21.5
7	2.090	0.210	0.523	0.31	0.210	0.55	0.98	0.63	9.75
7	2.198	0.210	0.523	0.31	0.318	0.53	1.2	0.71	12.25
7	2.303	0.210	0.523	0.31	0.423	0.54	1.4	0.79	14.75
7	2.408	0.210	0.523	0.31	0.528	0.56	1.6	0.87	17.25
7	2.513	0.210	0.523	0.31	0.633	0.58	1.9	0.96	19.75
7	3.313	0.375	0.425	0.35	0.313	1.01	4.8	2.1	15.6
7	3.350	0.375	0.425	0.35	0.350	0.99	5.1	2.2	16.5
7	3.438	0.375	0.425	0.35	0.438	0.96	5.7	2.3	18.6
7	3.450	0.475	0.525	0.35	0.450	1.05	6.7	2.8	20.9
7	3.500	0.475	0.525	0.35	0.500	1.05	7.1	2.9	22.1
7	3.550	0.475	0.525	0.35	0.550	1.04	7.5	3.0	23.3
8	2.260	0.220	0.560	0.32	0.220	0.58	1.3	0.79	11.25
8	2.347	0.220	0.560	0.32	0.307	0.56	1.6	0.87	13.75
8	2.439	0.220	0.560	0.32	0.399	0.56	1.8	0.95	16.25
8	2.530	0.220	0.560	0.32	0.490	0.57	2.0	1.0	18.75
8	2.622	0.220	0.560	0.32	0.582	0.59	2.3	1.1	21.25
8	3.415	0.475	0.525	0.35	0.415	0.99	6.9	2.9	21.5
8	3.500	0.475	0.525	0.35	0.500	0.99	7.4	3.0	23.8
8	3.550	0.475	0.525	0.35	0.550	0.98	7.8	3.0	25.2
8	3.600	0.475	0.525	0.35	0.600	0.98	8.2	3.1	26.5

TABLE XLII.

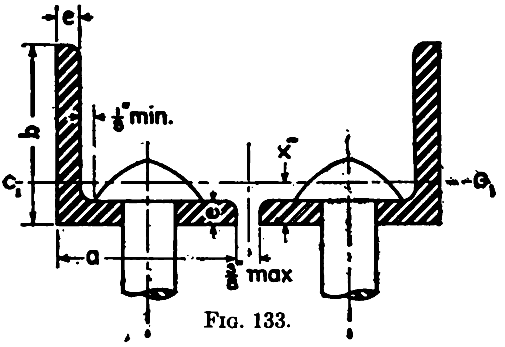


FIG. 133.

PROPERTIES OF STANDARD STRUCTURAL ANGLES

for use in Boiler Staying.

Equal legs.

$a \times b$ Dimensions of angle.	e Thickness of legs.	Radius of fillet.	x_1 Distance to C. G.	$I_{c.g.}$	$\frac{I_{c.g.}}{b-x_1}$.	Weight per foot.
Ins.	Ins.	Ins.	Ins.	Inch units.	Inch units.	Lbs.
3×3	$\frac{1}{4}$	$\frac{5}{16}$	0.84	1.2	0.58	4.9
3×3	$\frac{5}{16}$	$\frac{5}{16}$	0.87	1.5	0.71	6.1
3×3	$\frac{3}{8}$	$\frac{5}{16}$	0.89	1.8	0.83	7.2
3×3	$\frac{7}{16}$	$\frac{5}{16}$	0.91	2.0	0.95	8.3
3×3	$\frac{1}{2}$	$\frac{5}{16}$	0.93	2.2	1.1	9.4
3½×3½	$\frac{5}{16}$	$\frac{3}{8}$	0.99	2.5	0.98	7.2
3½×3½	$\frac{3}{8}$	$\frac{3}{8}$	1.01	2.9	1.2	8.5
3½×3½	$\frac{7}{16}$	$\frac{3}{8}$	1.04	3.3	1.3	9.8
3½×3½	$\frac{1}{2}$	$\frac{3}{8}$	1.06	3.6	1.5	11.1
3½×3½	$\frac{9}{16}$	$\frac{3}{8}$	1.08	4.0	1.6	12.4
3½×3½	$\frac{5}{8}$	$\frac{3}{8}$	1.10	4.3	1.8	13.6
4×4	$\frac{5}{16}$	$\frac{3}{8}$	1.12	3.7	1.3	8.2
4×4	$\frac{3}{8}$	$\frac{3}{8}$	1.14	4.4	1.5	9.8
4×4	$\frac{7}{16}$	$\frac{3}{8}$	1.16	5.0	1.8	11.3
4×4	$\frac{1}{2}$	$\frac{3}{8}$	1.18	5.6	2.0	12.8
4×4	$\frac{9}{16}$	$\frac{3}{8}$	1.21	6.1	2.2	14.3
4×4	$\frac{5}{8}$	$\frac{3}{8}$	1.23	6.7	2.4	15.7
4×4	$\frac{11}{16}$	$\frac{3}{8}$	1.25	7.2	2.6	17.1
4×4	$\frac{3}{4}$	$\frac{3}{8}$	1.27	7.7	2.8	18.5
6×6	$\frac{3}{8}$	$\frac{1}{2}$	1.64	15.4	3.5	14.9
6×6	$\frac{7}{16}$	$\frac{1}{2}$	1.66	17.7	4.1	17.2
6×6	$\frac{1}{2}$	$\frac{1}{2}$	1.68	19.9	4.6	19.6
6×6	$\frac{9}{16}$	$\frac{1}{2}$	1.71	22.1	5.1	21.9
6×6	$\frac{5}{8}$	$\frac{1}{2}$	1.73	24.2	5.7	24.2
6×6	$\frac{11}{16}$	$\frac{1}{2}$	1.75	26.2	6.2	26.5
6×6	$\frac{3}{4}$	$\frac{1}{2}$	1.78	28.2	6.7	28.7
6×6	$\frac{13}{16}$	$\frac{1}{2}$	1.80	30.1	7.2	31.0
6×6	$\frac{7}{8}$	$\frac{1}{2}$	1.82	31.9	7.6	33.1
6×6	$\frac{15}{16}$	$\frac{1}{2}$	1.84	33.7	8.1	35.3
6×6	1	$\frac{1}{2}$	1.86	35.5	8.6	37.4

TABLE XLIII.

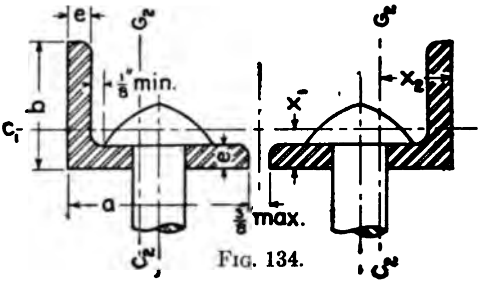


Fig. 134.

PROPERTIES OF STANDARD STRUCTURAL ANGLES

for use in Boiler Staying.

Unequal legs.

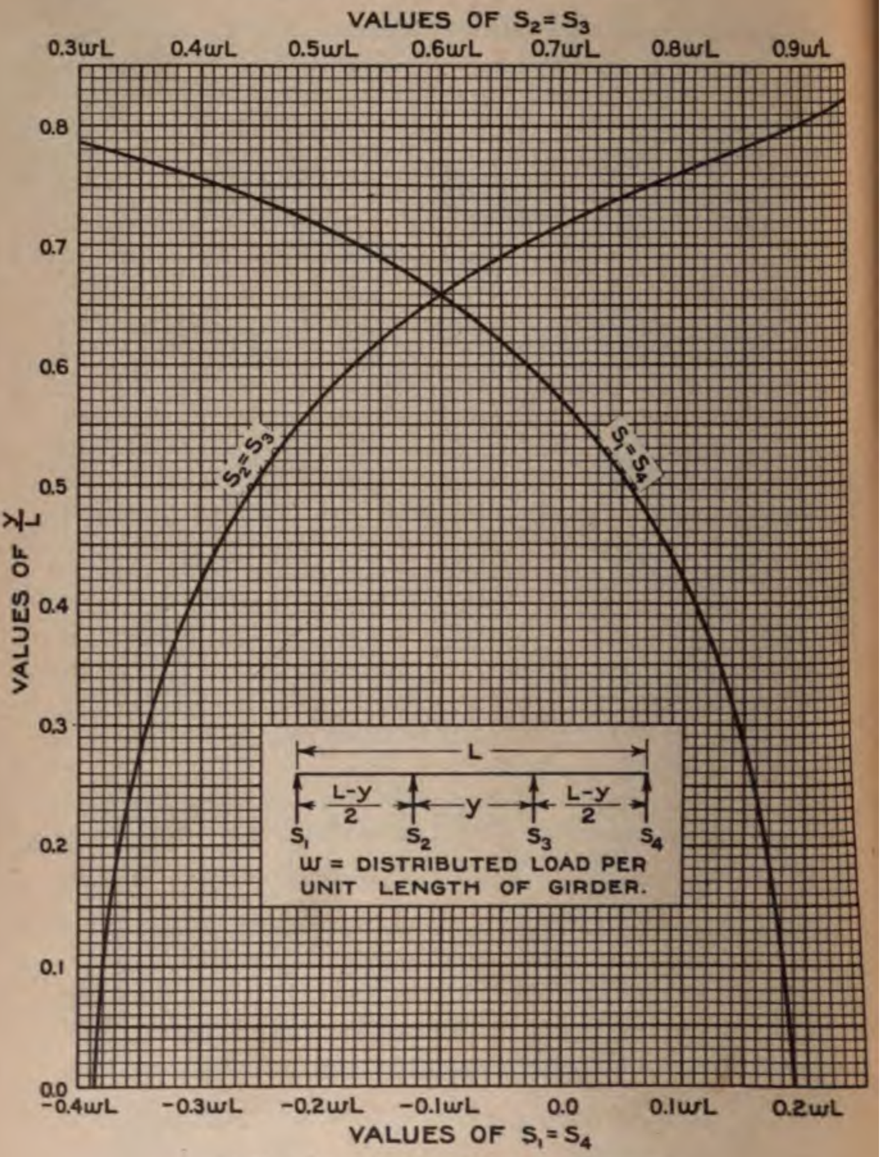
$a \times b$ Dimensions of angle.	e Thickness of legs.	Radius of fillet.	x_1 Distance to C. G.	$I_{C.G.\,1-1}$	$\frac{I_{C.G.\,1-1}}{b-x_1}$	x_2 Distance to C. G.	$I_{C.G.\,2-2}$	$\frac{I_{C.G.\,2-2}}{b-x_2}$	Weight per foot.
Ins.	Ins.	Ins.	Ins.	Inch units.	Inch units.	Ins.	Inch units.	Inch units.	Lbs.
3×2½	¼	5/16	0.66	0.74	0.66	0.91	1.2	0.56	4.5
3×2½	5/16	5/16	0.68	0.90	0.68	0.93	1.4	0.69	5.6
3×2½	⅜	5/16	0.71	1.0	0.71	0.96	1.7	0.81	6.6
3×2½	7/16	5/16	0.73	1.2	0.73	0.98	1.9	0.93	7.6
3½×2½	¼	5/16	0.61	0.78	0.41	1.11	1.8	0.75	4.9
3½×2½	5/16	5/16	0.64	0.94	0.50	1.14	2.2	0.93	6.1
3½×2½	⅜	5/16	0.66	1.1	0.59	1.16	2.6	1.1	7.2
3½×2½	7/16	5/16	0.68	1.2	0.68	1.18	2.9	1.3	8.3
3½×2½	½	5/16	0.70	1.4	0.76	1.20	3.2	1.4	9.4
3½×3	5/16	⅜	0.81	1.6	0.72	1.06	2.3	0.96	6.6
3½×3	⅜	⅜	0.83	1.8	0.85	1.08	2.7	1.1	7.9
3½×3	7/16	⅜	0.85	2.1	0.98	1.10	3.1	1.3	9.1
3½×3	½	⅜	0.88	2.3	1.1	1.13	3.5	1.5	10.2
3½×3	9/16	⅜	0.90	2.5	1.2	1.15	3.8	1.6	11.4
4×3	5/16	⅜	0.76	1.7	0.74	1.26	3.4	1.2	7.2
4×3	⅜	⅜	0.78	1.9	0.87	1.28	4.0	1.5	8.5
4×3	7/16	⅜	0.80	2.2	1.0	1.30	4.5	1.7	9.8
4×3	½	⅜	0.83	2.4	1.1	1.33	5.0	1.9	11.1
4×3	9/16	⅜	0.85	2.7	1.2	1.35	5.6	2.1	12.4
4×3	⅝	⅜	0.87	2.9	1.4	1.37	6.0	2.3	13.6
5×3	5/16	⅜	0.68	1.8	0.75	1.68	6.3	1.9	8.2
5×3	⅜	⅜	0.70	2.0	0.89	1.70	7.4	2.2	9.8
5×3	7/16	⅜	0.73	2.3	1.0	1.73	8.4	2.6	11.3
5×3	½	⅜	0.75	2.6	1.1	1.75	9.5	2.9	12.8
5×3	9/16	⅜	0.77	2.8	1.3	1.77	10.4	3.2	14.3
5×3	⅝	⅜	0.80	3.1	1.4	1.80	11.4	3.5	15.7
5×3	11/16	⅜	0.82	3.3	1.5	1.82	12.3	3.9	17.1
5×3½	5/16	7/16	0.84	2.7	1.0	1.59	6.6	1.9	8.7
5×3½	⅜	7/16	0.86	3.2	1.2	1.61	7.8	2.3	10.4
5×3½	7/16	7/16	0.88	3.6	1.4	1.63	8.9	2.6	12.0
5×3½	½	7/16	0.91	4.0	1.6	1.66	10.0	3.0	13.6
5×3½	9/16	7/16	0.93	4.4	1.7	1.68	11.0	3.3	15.2
5×3½	⅝	7/16	0.95	4.8	1.9	1.70	12.0	3.7	16.8
5×3½	11/16	7/16	0.97	5.2	2.1	1.72	13.0	4.0	18.3
5×3½	¾	7/16	1.00	5.6	2.2	1.75	13.9	4.3	19.8

pressures two rods only need be employed upon the lower row. The channels thus riveted and supported may be considered to constitute uniformly loaded continuous girders extending across the boiler heads. They are supported at four or more points. The stiff flange at the sides forms the outer pair of supports and consequently the girder should be made long enough to reach as far toward the shell as the flat portion of the head will permit. The stay rods with their check nuts constitute the remaining supports.

Having assumed the center space 16 ins. on the upper row, the outer spans of this continuous girder may be measured from the drawing. The lower stay rods are spaced symmetrically, the distance between them varying from 14 ins. to 20 ins., depending upon the pressure and amount of plate to be stayed. With three rods on the lower row the spacing rarely exceeds 16 ins. The four spans on the lower continuous girder may, therefore, be determined from the drawing. The calculation of the supporting forces and bending moments of a continuous girder is somewhat complex, involving an extended application of the beam theory. The three assumptions that the supports are in line, that the girder is of uniform section throughout and that the load is uniformly distributed, so simplifies the matter that plots like the accompanying ones, Figs. 135 and 136, can easily be constructed from which may be read the desired quantities.

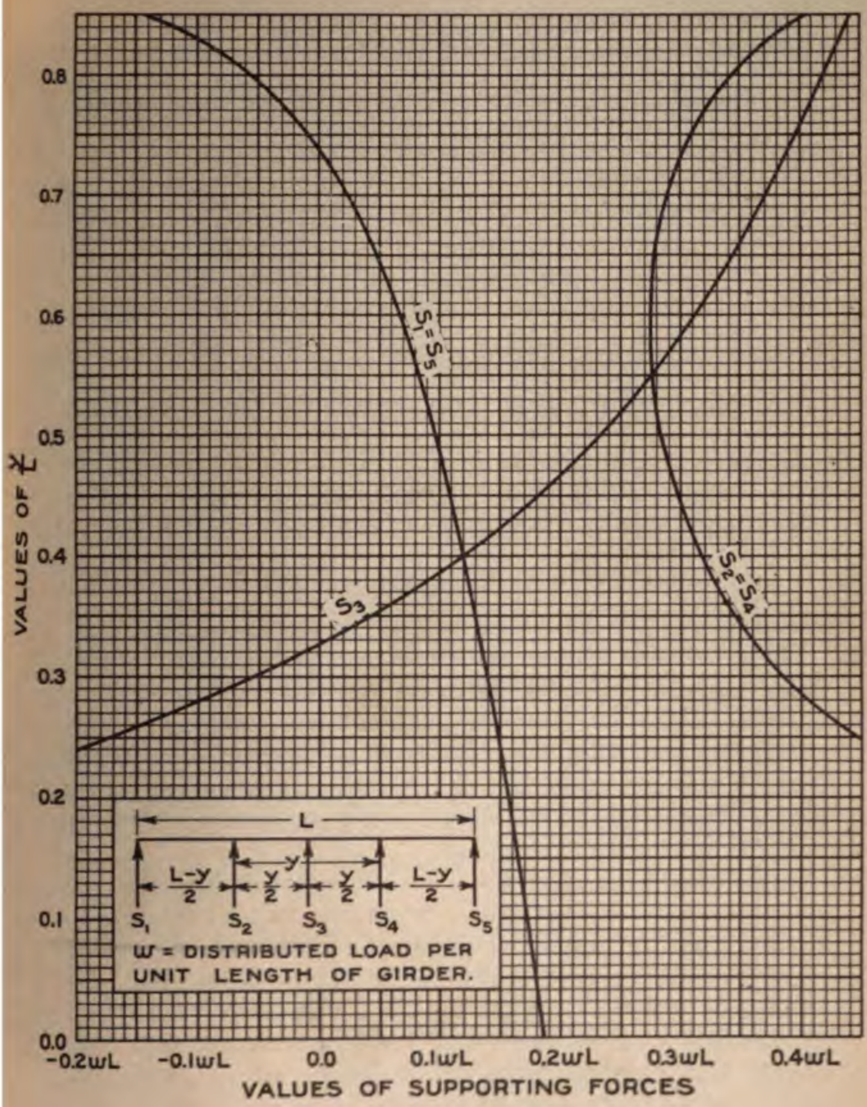
In the plot of the four-support girder, Fig. 135, the central span y is taken as a variable, passing in value from 0 to L. It is easily seen that the ratio $\frac{y}{L}$ as fixed by the particular problem at hand enables the values of S_1, S_2, S_3 and S_4 to be determined in terms of w the load per unit of length and L the total length of the girder.

The assumption that the supports are in line brings about the condition that, with large values of y, the outer supporting forces become negative. In other words, for values of $\frac{y}{L}$ greater than 0.57 the outer ends of the girder have to be held down in order to keep the supports in line. If they are not held down the girder becomes a simple overhung beam resting on two supports. It will be noted that, for $\frac{y}{L} = 0.57$, the value of $S_2 = S_3$ is $\frac{wL^2}{2}$ which naturally follows.



PLOT OF SUPPORTING FORCES
FOR
CONTINUOUS GIRDER

Fig. 135.



PLOT OF SUPPORTING FORCES
FOR
CONTINUOUS GIRDER

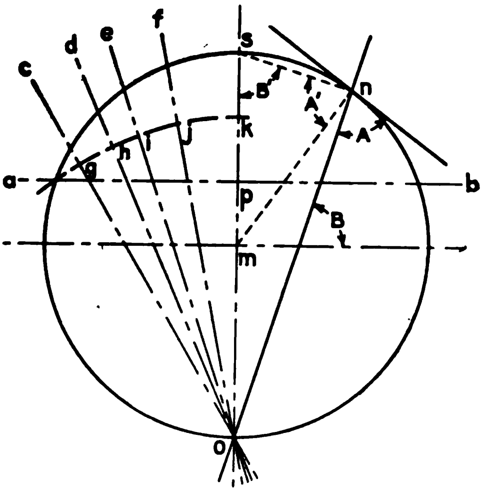
Fig. 136.

In the plot for a five-support girder, Fig. 136, the sum of the two equal central spans is made the variable. Here for a value of $\frac{y}{L} = 0.554$ the supporting forces S_2, S_3 and S_4 are equal, a desirable condition in stay rods. This same condition may be expressed by saying that when each of the equal central spans is 27.7 per cent of the girder length, the values of S_2, S_3 and S_4 are equal. Again it will be noted that for values of $\frac{y}{L}$ greater than 0.74 the outer supporting forces, $S_1 = S_5$, become negative. Also for $\frac{y}{L} = 0.326$ the central supporting force S_3 becomes zero and for all greater values of $\frac{y}{L}$ the center of the girder must be held down in order to keep the supports in line. If the center is not held down the case reverts to the previous one, a four-support girder with central span y. To confirm this, it will be noted that for $\frac{y}{L} = 0.33$ the values on the four-support girder, $S_2 = S_3$, are the same as $S_2 = S_4$ on the five-support girder, namely, $0.36\ wL$.

The distributed load w, to be used per unit of length, is that which naturally belongs to the girder in question. It is found as indicated in Fig. 131. Lines *mn* and *pq* are drawn midway between rows of rivets to indicate the boundaries of plate supported. The height of this area multiplied by the working pressure per square inch as represented by the shaded rectangle gives the load per unit of length on the lower girder.

In the case of the upper girder the division of load between the top row of rivets and the shell is determined by the method indicated in Fig. 137. A series of lines should be drawn forming equal angles with the upper rivet row and a tangent to the shell. By pricking off halfway from the rivet line to the shell along these mean radials, a curved load line *st*, Fig. 131, is determined. This varying load is approximated by the horizontal line *uv* and the shaded load rectangle is found as before.

Many cases occur where it is desirable to divide a loaded area between straight and curved lines. An easy method for doing this is shown in Fig. 137. In staying problems when it is desired to allot an unsupported area *aps* between a row of stay rivets *ab* and the shell, the following method is believed to give a fair sub-division. Let *ab* represent the straight line, the curve being the upper arc of the circle. A series of lines *oc*, *od*, *oe* and *of* will form, respectively, equal angles with the chord *ab* and a tangent at their points of intersection with the arc. If a series of points *ghij* be pricked off halfway between the chord and arc, the locus *gk* will divide the area fairly between the straight and curved lines. The lines *oc*, *od*, *oe* and *of* are called mean radials to the chord and arc. Referring to Fig. 137, the proof of their relation to the tangents is as follows:



METHOD OF DRAWING
MEAN RADIALS
FIG. 137.

Drawing the dotted construction lines as shown,

Angle B' = Angle A'

since triangle *mns* is isosceles.

But Angle A = Angle A'

and Angle B = Angle B'

since their sides are mutually perpendicular.

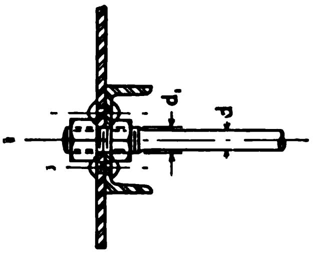
Hence Angle A = Angle B.

Q.E.D.

Lines similar to *on* then, forming equal angles with the chord and tangent, are fair ones on which to lay off the dividing line *gk*.

Having determined the several loads the diameter of the stay rods should next be found. It is not customary in boiler practice to use stay rods of different diameters in the same boiler. Hence the largest supporting force should be selected and the shank diameter of all the through rods based upon it. Stay rods are always upset at the threaded end so that the diameter at the root of the thread is substantially greater than at the shank. Table XLIV accompanied by Fig. 138 gives the usual proportions of

TABLE XLIV.



DETAIL OF THROUGH ROD

Fig. 138.

PROPORTIONS OF UPSET STAY RODS.

Rods finished as below are stronger at the thread root than at the shank.

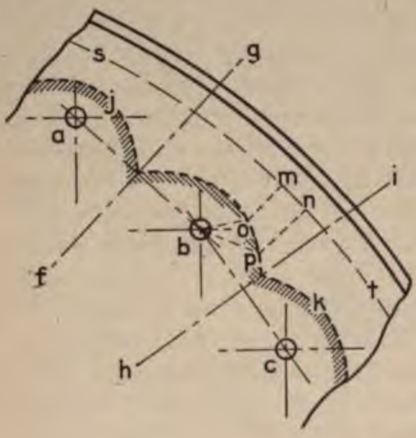
Tensile strength = 55,000 lbs. per sq. in. Factory of safety = 7.

d Diam. of shank.	d_1 Diam. of upset.	Area of shank.	Threads per inch.	Working load.	Short diam. of hex. nut.
Ins.	Ins.	Sq. ins.		Lbs.	Ins.
1	1¼	0.79	7	6,200	2
1⅛	1⅜	0.99	6	7,800	2³⁄₁₆
1¼	1½	1.23	6	9,700	2⅜
1⅜	1¾	1.48	5	11,600	2¾
1½	1⅞	1.77	5	13,900	2¹⁵⁄₁₆
1⅝	2	2.07	4½	16,300	3⅛
1¾	2⅛	2.41	4½	18,900	3½
1⅞	2¼	2.76	4½	21,700	3½
2	2⅜	3.14	4	24,700	3⅞
2⅛	2½	3.55	4	27,900	3⅞
2¼	2⅝	3.98	4	31,300	4¼
2⅜	2¾	4.43	4	34,800	4¼
2½	2⅞	4.91	3½	38,600	4⅝
2⅝	3	5.41	3½	42,500	4⅝
2¾	3⅛	5.94	3½	46,700	5
2⅞	3¼	6.49	3½	51,000	5
3	3⅜	7.07	3½	55,600	5⅜

upset stay rods and the loads which they will sustain. The latter are based upon a tensile strength of 55,000 lbs. per sq. in. with a factor of safety of seven. As soon as the rod loads are determined the size can be selected directly from Table XLIV by inspection. Before accepting a shank diameter it is well to be sure that corrosion to the depth of $\frac{1}{16}$ in. over the rod surface will not render it liable to be strained beyond the elastic limit at the test pressure.

V. Final Test for Fibre Stress in Channels or Angles. — Having drawn the stay rods in position with standard nuts upon their ends, Fig. 131, it next remains to space the rivets along their respective rows. Since the locality around the check nuts possesses great inherent strength it would hardly be fair to space the rivets uniformly between stay rod centers. Hence a line xx' is drawn across each check nut at a distance from its center equal to one-half the long radius of the hexagon and the unsupported plate is supposed to begin at this point. The rivets are then spaced equally between the lines xx', as shown. In order to insure as flat plate as possible for the nuts to rest against, four rivets must always be placed symmetrically around each stay rod. An additional rivet on each row is then spaced in when necessary halfway between the above mentioned rivets and the flange curve. With very high pressures two rivets may be needed. The manner of determining the load upon each rivet is indicated by the shaded areas in Fig. 131. In general, a line is drawn halfway between each rivet and its natural neighbor. The areas of the figures thus determined, when multiplied by the working pressure, give the loads sustained by the rivets. In no case should these areas exceed that given in Table XL for the rivet and pressure at hand.

The determination of the areas supported by the rivets near the flange curve is attended with some question but the method outlined in Fig. 139 appears to be a fair one. When a series of rivets *abc* are irregularly arranged in proximity to a stiff curve *st* the rivet centers should first be connected by the straight lines shown and the perpendiculars *fg*, *hi*, etc., erected at their central points. These lines form the boundaries of the areas in a radial direction. Between the rivets and the circumference, the dotted line *jk* is so drawn as to form the locus of points equidistant from the successive rivet centers and the stiff line *st*. That is, *bo* equals *om* and *bp* equals *pn*. Such a line, therefore, together with the perpendiculars, forms the boundary of the shaded area supported by the rivet *b.* This principle is applicable to many other cases where the load sustained by stays is in question. The method was employed in Fig. 131 and the value of the results is there plainly evident. When the resulting figures are quadrilaterals of some regularity, the areas may be found approximately by direct measurement. When very irregular, a planimeter may be required to gain a fair value of their extent.



DETERMINATION
OF
RIVET LOADS
Fig. 139.

It is found from the theory of continuous girders that the maximum bending moment usually occurs over a support, while that in the spans is much less in amount. With check nuts and stiff plate reinforced by the channel itself, the bending moments at the supported points in the continuous girders at hand are largely eliminated and those in the spans become the more important ones. Also the fact that the supports may not be in line, due to differences of expansion in the rods and shell, tends seriously to disarrange the theoretical calculation of the bending moments. Hence at this point in the design the continuous girder assumption is dismissed and a conservative check calculation made, considering the channels between stay rods as beams merely supported at the ends and subjected to loads concentrated at the various points where the rivets take hold. The approximate length of beam shown in Fig. 131 is assumed and, from the area of plate which each rivet supports, the concentrated loads are calculated. The reaches of channel from stay rods to shell are rarely overstrained. With the same channel used throughout in the staying, the most severely loaded beam should be selected and calculated for fibre stress. The working modulus of rupture of the angles or channels in this calculation should not exceed 14,000 lbs. per sq. in. on the compression side, indicating a factor of safety of approximately four. The maximum bending moment divided by this quantity gives the section modulus which may be compared directly with the values given in Tables XLI, XLII and XLIII.

The above procedure completes the usual calculation for girder stays and, if carefully pursued, will give a safe and satisfactory design.

(*b*) *Diagonal Braces.* — The design of diagonal stays is carried out in somewhat the same order as that followed for through rods. Figs. 140, 141 and 142 show the three general types of diagonal stays at present most used. The Scully brace is made from one piece of soft steel pressed to shape in a forging machine, no welds being allowed. For general use the sizes given in Table XLV are obtainable.

TABLE XLV.

PROPORTIONS OF SCULLY BOILER BRACES.

Diam. of body.	Size of head end.	Size of shell end.	Area of body.
Ins.	Ins.	Ins.	Sq. ins.
$1\frac{1}{8}$	$6\frac{1}{2}\times2\frac{1}{4}\times\frac{7}{16}$	$8\frac{1}{2}\times3\frac{1}{4}\times\frac{7}{16}$	0.994
$1\frac{3}{16}$	$6\frac{1}{2}\times2\frac{1}{4}\times\frac{1}{2}$	$8\frac{1}{2}\times3\frac{1}{4}\times\frac{1}{2}$	1.108
$1\frac{1}{4}$	$6\frac{1}{2}\times2\frac{1}{4}\times\frac{5}{8}$	$8\frac{1}{2}\times3\frac{1}{4}\times\frac{9}{16}$	1.227

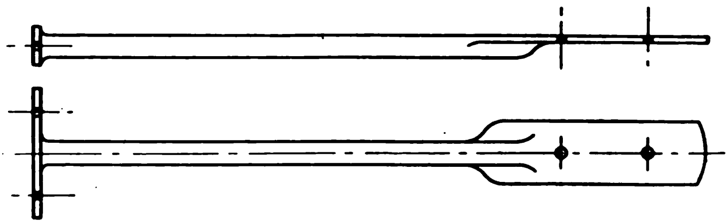
Length may vary from 24 ins. to 108 ins. inclusive by increments of six inches.

The Huston brace is made of folded weldless steel plate. At its head end it is well reinforced by wide fillets and at its shell end by raised ribs along the sides. The thickness of metal from which this brace is made ranges from $\frac{3}{8}$ in. to $\frac{1}{2}$ in., depending upon the pressure in the boiler in which it is used. The lengths range from 24 ins. to 78 ins. inclusive by increments of 6 ins.

The McGregor brace is also made of folded metal but differs from the preceding one in the formation of its head. The latter is split and bent to a T-shape. Weldless material, varying in thickness from $\frac{3}{8}$ to $\frac{1}{2}$ in., is used throughout. The shank areas are as follows:

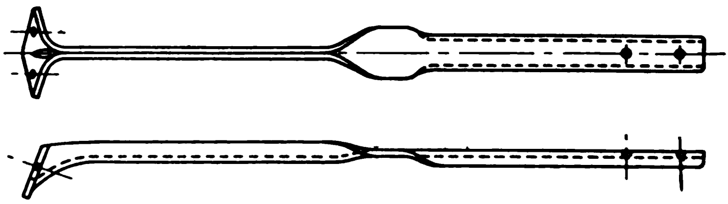
For $\frac{3}{8}$ in. thickness, 1.13 sq. ins.
For $\frac{7}{16}$ in. thickness, 1.31 sq. ins.
For $\frac{1}{2}$ in. thickness, 1.63 sq. ins.

The lengths are the same as for the Huston brace.



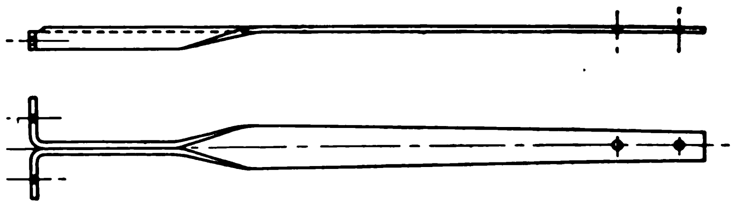
SCULLY BRACE

Fig. 140.



HUSTON BRACE

Fig. 141.



McGREGOR BRACE

Fig. 142.

DIAGONAL BRACES

All the above diagonal stays are designed to seat two $\frac{7}{8}$ in. rivets spaced 4 ins. apart at both ends and are supposed to contain enough material in their shanks to sustain the maximum rivet loads with a factor of safety of seven or eight. It is evident that the brace tension will increase with the slope of the stay. The shell angle is generally limited to 20 degrees with an extreme maximum of 30 degrees. The distance between the rivets at the shell end may be increased to 6 ins. if necessary when the brace is anchored to the rivets of a longitudinal seam. In any event the first rivet in the shell anchorage should come as near the bend in the brace as it can be driven so as to remove as effectually as possible the tendency for the brace to straighten out under pressure. In general a rivet in the shell end of a brace should not come nearer than 3 ins. to a ring seam. Diagonal braces are furnished by the manufacturers with the heads and shanks forged straight as in Figs. 140, 141 and 142. The head of the Huston brace is given a slight angle by its method of manufacture. When ready for insertion in the boiler the brace is heated and forged to the exact shape required. The holes, previously drilled in the boiler head and shell, are then "scribed" through upon the brace and accurately located so as to secure as snug a fit as possible.

Fig. 143 shows the method of arranging diagonal braces in the upper portion of the tube sheet of a horizontal cylindrical boiler. The spacing of the braces must be such as to load the rivets as equally as possible. Since this form of staying is of itself somewhat more flexible than through rods, the limiting conditions to secure flexibility need not be so severe. The following limits should be observed:

(1) *No rivet shall be less than 3 ins. from the shell.*

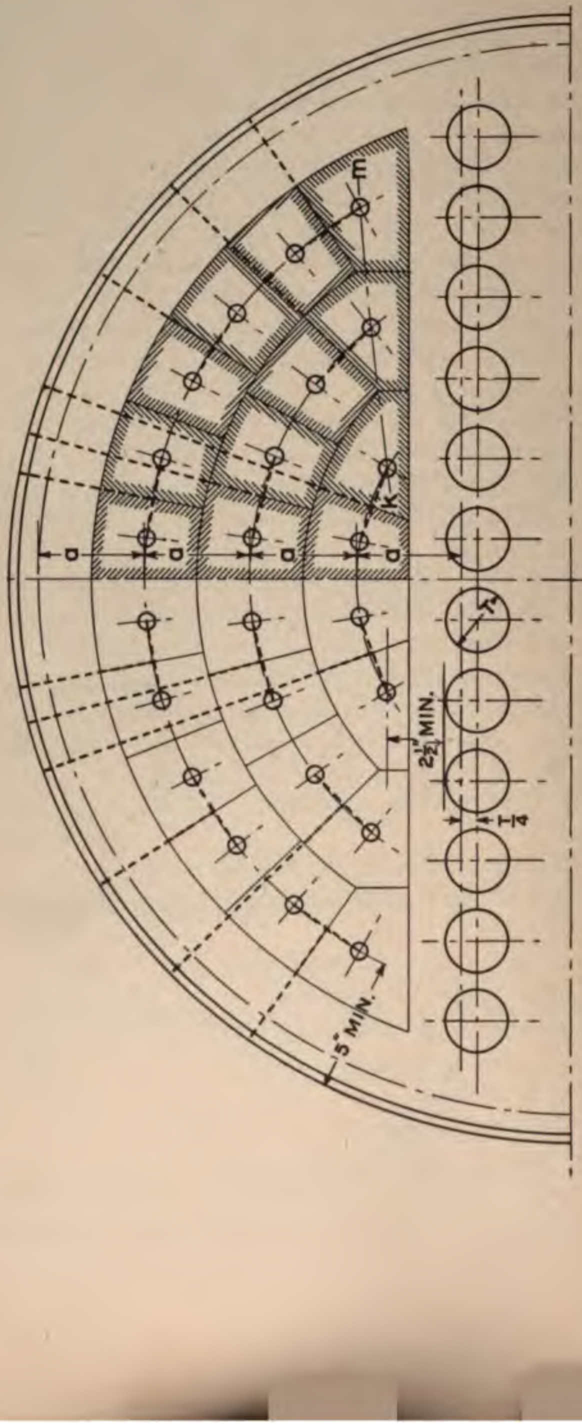
(2) *No rivet shall be less than $2\frac{1}{2}$ ins. from a neighboring rivet or the top of tubes.*

(3) *No stay rod head shall be less than 5 ins. from the shell.*

The following is the method of procedure:

I. *Maximum spacing as regards plate.*
II. *Maximum spacing as regards rivet.*
III. *Arrangement of braces.*
IV. *Test for tension in braces.*
V. *Test for stresses in anchor rivets.*

I and II are carried out by tabular reference as described under through rods, and form the basis of the arrangement of the braces.



TUBE SHEET STAYING

DIAGONAL BRACES

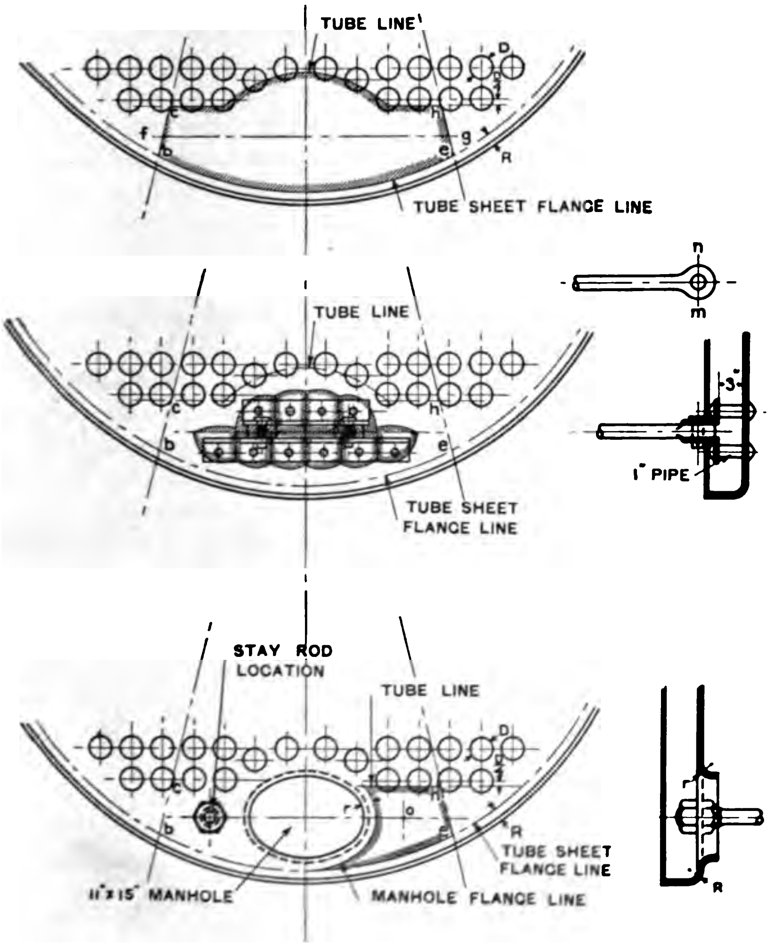
Fig. 143.

III. Arrangement of Braces. — Having determined the area of plate to be stayed, as in the previous case, an estimate may be made of the number of layers or zones of braces required. To do this divide the total height of the segment to be stayed by the maximum sheet spacing as denoted by Table XXXIX. The next integer above the quotient obtained will indicate the number of zones in the staying. Thus suppose the maximum sheet spacing from Table XXXIX to be 5.40 ins. Then, if the height of the segment to be stayed is 18 ins., there will be four zones and three layers of braces will be required. Hence the height of the segment should be divided graphically into four concentric zones as indicated in Fig. 143. These lines indicate the circular rows of brace rivets. Had only two layers of braces been needed the segment would have been divided into three concentric zones. Diagonal stays should next be spaced along the several circles with a uniform distance of 4 ins. between rivets. This is a desirable arrangement since it gives uniform loads on all the rivets. The method, however, can rarely be carried out as directly as the above would indicate, inasmuch as the procedure is liable to result in an irregular boundary line *km* for the lower rivet centers. This line should be approximately straight, and by varying the distance from brace to brace this condition can be secured. The spacing of the braces in each zone, however, must be entirely uniform. The manner in which the area supported by each rivet is determined is clearly indicated in Fig. 143. It is sometimes desirable to employ the method of loci shown in Fig. 139, page 248, to determine the lower rivet loads. In order to bear its load in the most advantageous manner possible, each brace should reach back radially to the shell for support. Care must be taken that the braces do not thus conflict. A slight deviation from the radial position is permissible where braces are of necessity anchored to the rivets of longitudinal seams. The radii of the various rivet zones and the chordal distances between rivet centers should be indicated to the nearest $\frac{1}{16}$ inch.

IV. Test for Tension in Braces. — Having found a satisfactory spacing the maximum load for each brace should be determined. Since the braces slope the tension in the body will be much greater than the tube sheet load, increasing with the shell angle. It is not necessary to analyze the stress for each brace contained in the boiler. With a little care the brace under the severest conditions of loading may be selected and its analysis made to serve as a limit for all the other braces used. Graphical methods are the easiest to employ in this connection. A factor of safety of seven or eight should be secured on a basis of 60,000 lbs. per sq. in. ultimate tensile strength.

V. Test for Stresses in Anchor Rivets. — The rivets which anchor the brace to the shell and head should be calculated graphically for tension, shearing and bearing pressure, and large factors of safety secured in order to avoid excessive local stresses. No anchor rivet should approach within 3 ins. of the ring seam pitch line. When a brace must of necessity be anchored to a longitudinal seam, the rivets in the anchor pad should be made to conform in their spacing with those selected for the purpose in the joint.

Staying at Bottom of Tube Sheet. — When a manhole is employed in the front head of a horizontal boiler and the requisite number of tubes are omitted for its insertion, there is left a considerable area of unstayed plate in the rear tube sheet. Careful provision must be made for supporting this area, the location of which prohibits the use of large masses of metal. It is impossible to rivet channels or stiffeners of any kind directly to the tube sheet, since the combined thickness of metal would cause overheating and consequent burning. Fig. 144 shows a satisfactory method of arranging the stays at this point. To the rear tube sheet are riveted two heavy angles turned flange to flange and set off from the plate a minimum distance of 3 ins. by pipe thimbles or spacers inserted upon the rivets. The thimbles are simply 3 in. lengths of one inch pipe cut with squared ends. Rivets $\frac{15}{16}$ in. in diameter are generally used for this purpose to insure staunchness. The angles are set far enough apart to permit the insertion between them of a solid forged stay rod eye. The eye is made as wide as the diameter of the stay rod of which it is a part and is securely pinned to the angles. These stay rods reach to the front end of the boiler, spreading a little if necessary, pass through the tube sheet and are secured by nuts and check nuts. Their horizontal center line at the front tube sheet should not be lower than that of the manhole opening. The slight amount of spreading necessary does not hinder making a tight joint at the front sheet, a soft copper gasket being calked under the nut for the purpose. The following calculations should be made to insure sufficient strength.



STAYING BELOW TUBES IN HORIZONTAL BOILERS

FIG. 144.

I. Maximum spacing as regards plate.
II. Maximum spacing as regards rivet.
III. Calculation of size of rod.
IV. Calculation of eye.
V. Calculation of pin.
VI. Calculation of angles.

I and II. Plate and Rivet. — The first two of the above items are found as before by tabular reference and are the prime essentials in making the further calculations.

III. Size of Rod. — The area of plate to be stayed is determined as before by assuming the flange and tube lines as boundaries. The mean radial lines *bc* and *eh*, Fig. 144, should next be drawn in such a position that each is approximately equal to the maximum allowable plate spacing, as taken from Table XXXIX. Then the plate outside of these lines is stiff enough to take care of itself while the portion between them must be stayed. Next divide the unstayed area by planimeter or otherwise into two equal parts by the horizontal line *fg*. This is the center line of the rear staying system. Assuming a width of eye varying with the pressure in the boiler from $1\frac{1}{4}$ to $1\frac{3}{4}$ ins., lay out a tentative pair of angles from those given in the tables of standard sections. Four rivets are generally employed on the upper line and six on the lower, in order to reach the remote areas near the shell. The spacing on the lower line is generally somewhat greater than on the upper. From the temporary arrangement thus laid out, the actual rivet loads may be found by the principles described under Fig. 139. The irregular line bounding the rivet areas is the locus of points equally distant from the stiff lines and rivet centers. The area which each rivet supports should next be planimetered and the individual rivet loads at the working pressure calculated. The total of all the rivet loads is the load upon the stay rods. The threaded ends of the stay rods are based upon the proportions given in Table XLIV. From the data just calculated the size of rods may, therefore, be ascertained. It is desirable that the width of the forged eye shall be equal to or somewhat greater than the shank diameter of the rod. The width of eye should conform to the distance between angles previously assumed.

IV. Rod Eyes. — Without going into the theory of link eyes, it is a safe assumption, in cases of this character, to make the area on the section *mn*, Fig. 144, 50 per cent in excess of that employed in the rod shank itself. These proportions taken into account with the pin diameter will give a tentative size of eye.

V. Pins. — Taking the statical moments of the individual rivet loads about the vertical center line of the boiler, the proper point for the attachment of the stay rods to the angles may be located. Or if an integrator is at hand the center of gravity of the aggregate rivet areas may be found mechanically. A pin having an assumed diameter a little less than that of the stay rod should be used to complete the fastening of the stays in place. The pin should be calculated as a beam supported at the ends and loaded with a distributed load equal to the full stay rod load. With a suitable modulus of rupture, a factor of safety of 4 or 5 should be secured in the material forming the pin. The pin holes should be located as near the center of gravity of the angles as possible.

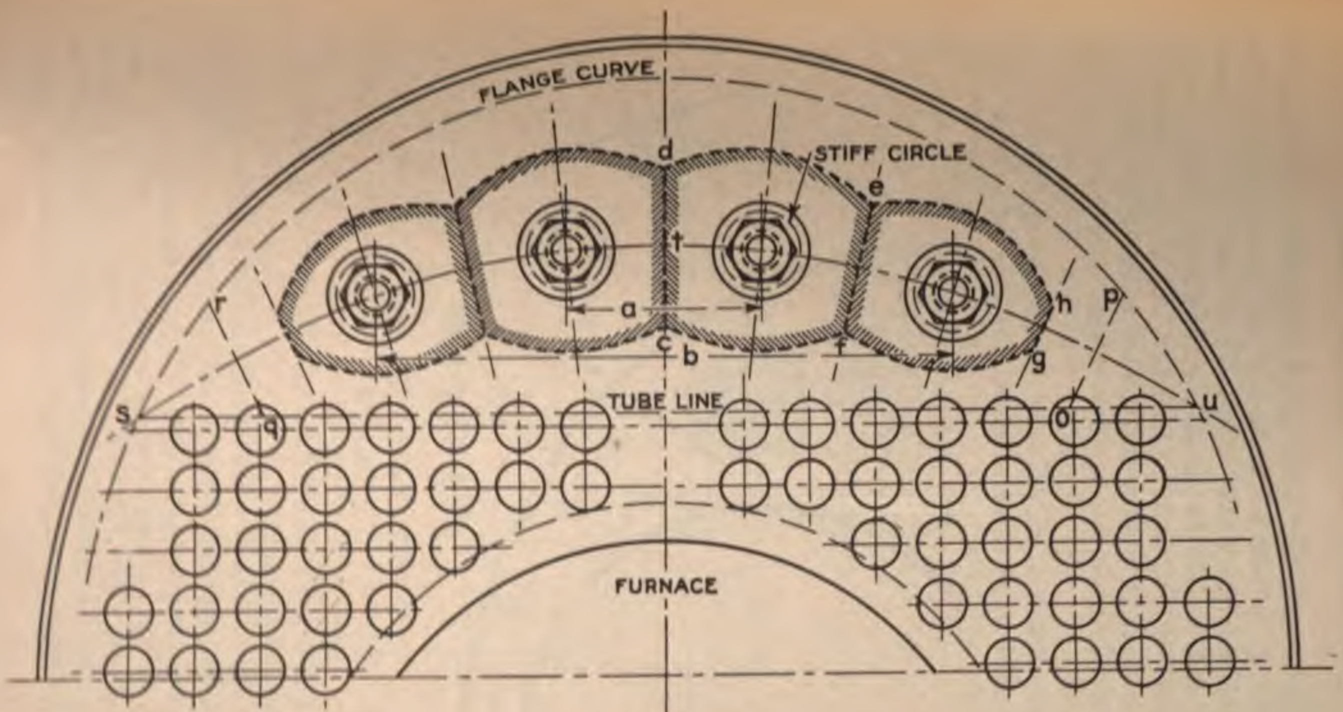
VI. Angles. — The upper angle should be calculated as a beam supported at the pin centers and loaded at the rivet centers. The lower angle constitutes an overhung beam and the bending effect between the pin centers is generally of minor importance. Its ends, however, should be calculated as cantilevers from the pin centers outward. As in the case of girder stays the factors of safety resulting from such calculations should not be less than 4 when based on an ultimate modulus of rupture of 55,000 lbs. per sq. in.

The location of the stay rod ends at the front tube sheet is illustrated by Fig. 144. The unstayed area, indicated by the shading, is determined precisely as before. The manhole area as far out as its flange line is considered inherently stiff. The center of gravity o of the shaded portion is therefore the correct position for the stay rod anchorage. As was remarked above, this point should be approximately upon the center line of the rear staying system. The exact location of o may be found by integrator or calculation.

Through Stay Rods in Scotch Boilers. — In Scotch boilers the removal of the upper portion of the tube sheet from direct contact with the fire makes it possible to replace the channel girders, previously described, with very heavy washers check-nutted upon the ends of the through rods. By this means the load upon a large area of the tube sheet is transferred to the stay rod. The thickness of the tube sheet itself may be made much greater in this type of boiler than in those where the heat of the fire is received by plate having behind it steam only. Fig. 145 shows the construction lines necessary in laying out this kind of staying. The unstayed plate is determined as before by drawing the flange curve and stiff line of the tubes. Across this area a series of mean radial lines should be drawn according to the principles described in Fig. 137, page 245. Pricking off halfway from the tube line to the flange curve, the medial line *stu* of the unstayed plate may be located. This curved line is the one upon which the stay rods are placed.

To determine the maximum allowable distance between stayed points, reference should be made to Table XXXIX, page 233. The lines *op* and *qr* are next located by experiment, each equal in length to the maximum plate spacing and so placed as to form mean radials to the flange curve and tube line. These lines, therefore, define the limit to which the plate is inherently stiff.

The washers used in this class of staying range generally from 6 to 8 ins. in diameter, varying by half inches. They are beveled slightly on the edge to facilitate calking and are generally of the same thickness as the tube sheet itself. Having assumed a tentative diameter of washer from the general extent of the surface to be stayed, a template to scale should be cut out of paper representing each washer to be used. As in the case of beaded tubes, it is here also a question as to how far the stiffness of the washer extends from its center. As a conservative estimate it is assumed that the washers are inherently stiff to a distance from their centers equal to the radius of the washer itself minus the thickness of the metal used. This "stiff circle" should be drawn upon each washer after which they may be laid upon the drawing along the medial line. The washer templates may next be placed by trial so that the reaches of plate between their stiff circles and the boundaries *op* and *qr*, Fig. 145, will be sensibly equal. The centers should finally be located some specified distance both horizontally and vertically from the center lines of the boiler. Lastly, assuming that the arrangement appears to be a satisfactory one, the method of loci described in Fig. 139 should be employed to find the area of plate stayed. Thus the shaded figures as shown are bounded by the loci of points equidistant from the flange line, the stiff circle of the washer and the tube line. Using a planimeter the extent of these areas may be found, and this multiplied by the working boiler pressure will give the normal load upon the

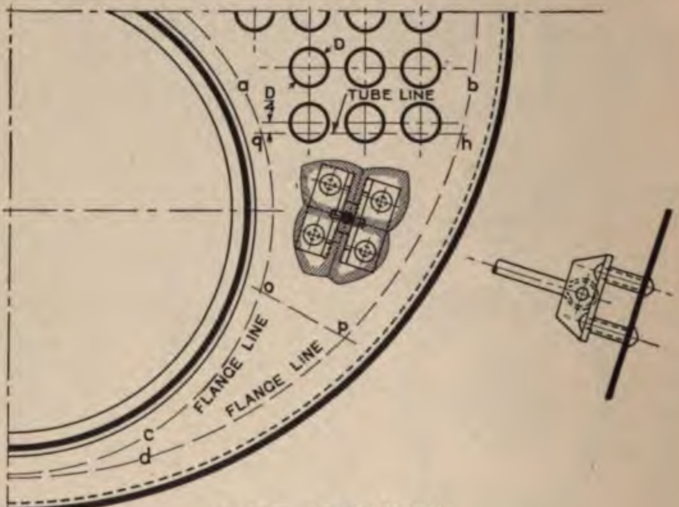


STAYING
FOR
DRY BACK SCOTCH BOILER

Fig. 145.

stay rods. Consulting Table XLIV, page 246, a diameter stay rod may be selected to fit the most severe case, and before, this diameter should be used for all rods above the tub The nut upon the outside should be full standard and that up the inside half standard in thickness. If in every case t reaches of plate between washers, tubes and shell fall below th maximum allowable spacing, the staying may be considere satisfactory.

The above method may also be extended to the case of larg Scotch boilers where several rows of stay rods are needed. Th



SIDE STAYING
IN
SCOTCH BOILERS

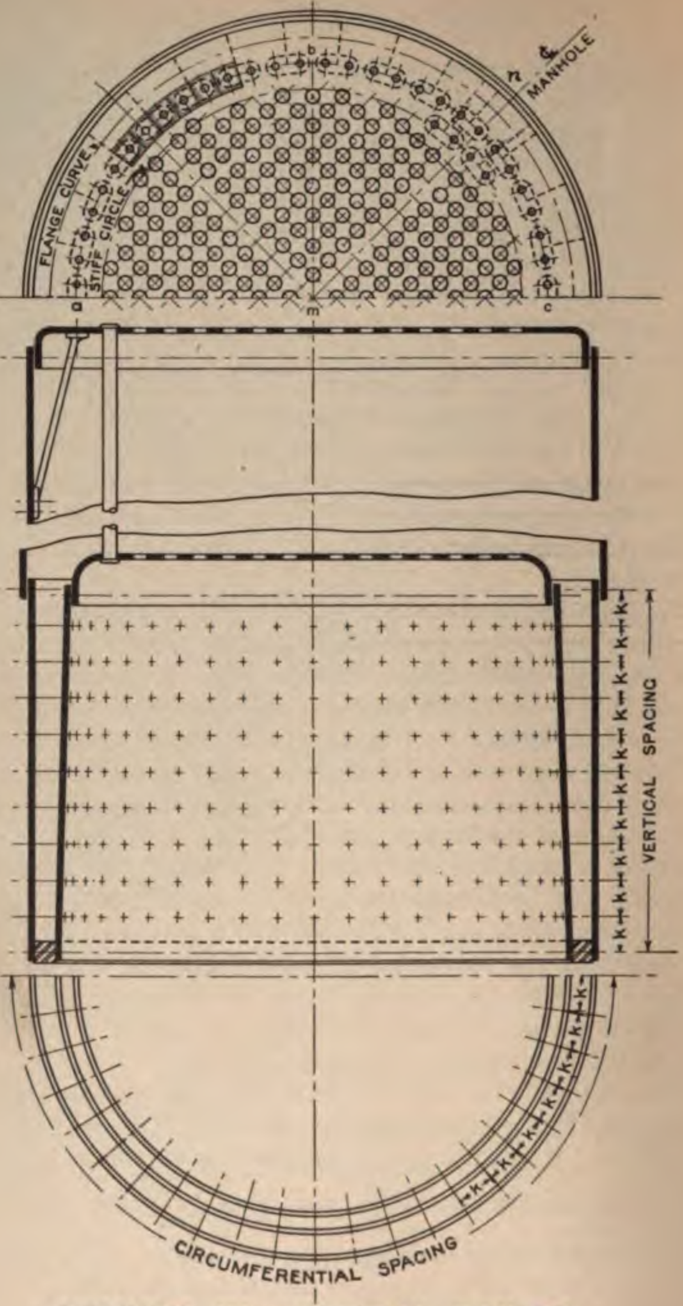
Fig. 146.

areas will differ in form from those found above but the principles applied will be the same.

Rear Tube Sheet Stays. — In Scotch boilers it is generally necessary to install large handholes in the front head at the sides of the furnace tubes near the shell. The omission of the tubes necessary for this purpose leaves a wide expanse of unstayed plate in the rear tube sheet. To determine whether this area must be stayed or not the procedure illustrated in Fig. 146 should be employed. Drawing the stiff line of the tubes and the flange curves for the furnace and shell, as previously described, the area *abcd* is determined. If the width of this figure, in a direction radial to the furnace and shell, is greater than the maximum allowable plate spacing in the head a stay rod is necessary. Let *op*, drawn so as to form equal angles with the two flange curves, represent the maximum plate spacing. The area *abop* must then be supported by the use of angles pinned to a stay rod in much the same manner as was previously described in Fig. 144, page 255. Four rivets are generally employed with 3 in. thimbles to permit circulation against the sheet. Having located the rivets as symmetrically as possible by trial the individual rivet loads are found by the method of loci described in relation to Fig. 139, page 248. With the sum of the rivet loads as the gross tension, a stay rod with forged eye should be designed, its forward end finding an anchorage in the reinforcing pad around the handhole opening. A slight amount of angularity is permissible where the rod joins the front tube sheet.

Staying in Vertical Boilers. — In vertical boilers there are two localities where stays are necessary, namely around the margin of the upper tube sheet and in the water legs. If the upper portion of the shell is contracted, the tubes may come near enough to the periphery of the tube sheet to stay it without additional assistance. In straight shell vertical boilers it is always necessary to use one or more zones of diagonal stays to strengthen the upper tube sheet. On account of the intense heat of the fire, stays cannot well be attached to the lower tube sheet. The necessity for their use is avoided, first, by flanging the lower tube sheet to a liberal radius and, second, by giving the furnace wall a gentle inward slope as it rises from the mud-ring. Fig. 147 shows the staying of the upper tube sheet and water leg as well as the manner in which the lower tube sheet joins the furnace wall. A slope of 2 or 3 ins. in the height of an ordinary furnace is allowable.

The unstayed plate at the circumference of the upper tube sheet is determined by the principles illustrated in Fig. 130, page 232. The flange curve is drawn where the flat plate begins and the tube circle at an approximate distance $\frac{D}{4}$ inside of the tube area. Naturally the latter determination is somewhat indefinite. If the unstayed area is wider than the maximum allowable plate spacing as determined from Table XXXIX, page 233, at least one row of stays will



STAYING IN VERTICAL BOILER

Fig. 147.

be needed. These should be uniformly spaced on the medial circle *abc* with 4 ins. between rivets. This spacing will rarely fit the medial circle. The distance between braces must, therefore, be varied to fit the case. With the diameter of the medial circle and the total number of braces given, the stays can be located without difficulty. For ease in laying out, the total number of braces should be a multiple of two or four. The determination of the shaded figures representing the area stayed per rivet is evident from the illustration. This area must not exceed that given in Table XL for the rivet and pressure in question. Where tubes are omitted locally to give easy access through the manhole as at *mn*, it is generally necessary to use a second row of stays. Their spacing and calculation follows the method outlined above.

Lastly the true rivet loads must be computed and the tension, shearing and bearing pressure kept low enough to avoid local distortion at the shell.

In the water legs stay bolts must be pitched, both vertically and circumferentially, so as to keep the furnace wall safe against collapse. The lowest ring seam in the outer shell should come approximately opposite the lower tube sheet ring seam. As was noted under water leg joints, Figs. 100 and 101, page 149, the strength of the furnace course in the outer shell is largely dependent upon the stay bolts. Above the region of the stay bolts, therefore, the longitudinal joint must at once pass to a type of high efficiency. The water space should be at least 3 ins. wide at the bottom and the mud-ring is generally made from $2\frac{1}{2}$ to 3 ins. in depth. Starting at the mud-ring seam, Fig. 147, a trial vertical spacing *kk* should be made with the maximum allowable plate spacing from Table XXXIX as a unit. To prevent burning, the furnace wall is generally restricted to $\frac{7}{16}$ in. or less in thickness. This quantity should be used in referring to the table. This uniform spacing, reduced if necessary, must reach the location of the lower tube sheet ring seam. If the levels of the ring seams in the lower tube sheet and shell are approximately the same, the arrangement may be considered satisfactory.

In large fire boxes the cylindrical form of the furnace is not considered to incorporate any added stiffness, since the external pressure tends to produce collapse. Hence the furnace wall is stayed circumferentially as if it were flat plate, the unit from Table XXXIX being the same as was used for the vertical spacing.

The stay bolts should be spaced on the outside circumference of the furnace wall at its largest diameter. Here again the total number of spaces should be a multiple of two or four, the pitch being slightly reduced to make such a number possible. The uniformity of stay bolt spacing should not be disturbed, but when necessary, the stay bolts may pass through more than one thickness of the outer plate. This makes it possible to use cover plates, pads and reinforcements on the outside of the shell without changing the position of the stay bolts. In the region of fire doors where there are stiff flanges present, the stay bolts may be occasionally displaced a little from their natural position.

It next remains to select a suitable size of stay bolt to bear the loads which the above spacing will impose upon them. The boundaries of the areas supported per stay bolt are located in the usual manner, and when multiplied by the working pressure, these areas will give the stay bolt loads. Table XLVI gives the dimensions usually adopted for stay bolts. Most railroads and boiler makers employ 12 threads per inch irrespective of the shank diameter, but 10 threads per inch have been used for the larger sizes. For U. S. Standard thread, the root diameter equals the shank diameter minus 1.30 times the pitch of the thread. If sharp V-thread is used the latter factor is 1.73 times the pitch. The following table has been calculated for U. S. standard threads:

TABLE XLVI.

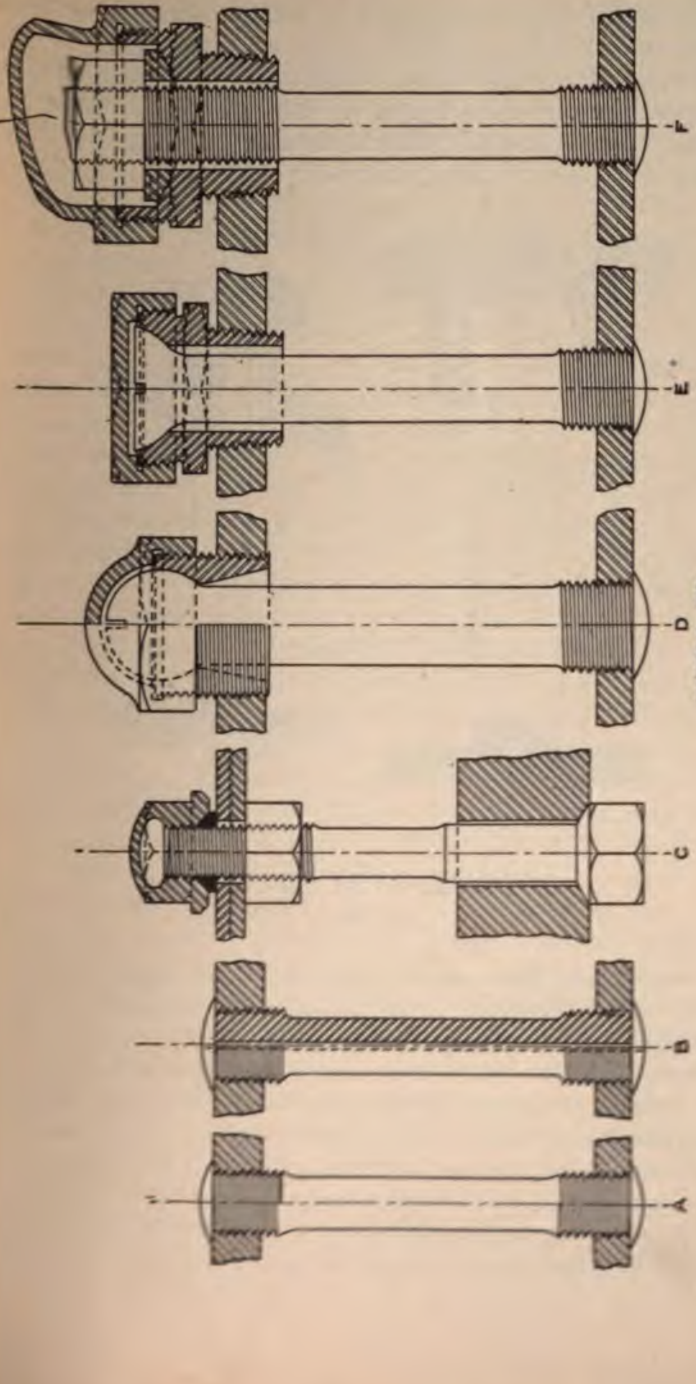
STANDARD PROPORTIONS
of
SOLID BOILER STAY BOLTS.

12 U. S. S. Threads per Inch.

Soft Steel, $f_t = 50{,}000$ lbs. per sq. in. Wrought Iron, $f_t = 45{,}000$ lbs. per sq. in.
Factor of Safety, 8.

Outside diam.	Area at thread root.	Strength, steel.	Strength, wrought iron.	Outside diam.	Area at thread root.	Strength, steel.	Strength, wrought iron.
In.	Sq. in.	Lbs.	Lbs.	Ins.	Sq. ins.	Lbs.	Lbs.
$\frac{1}{2}$	0.121	753	678	1	0.624	3903	3512
$\frac{5}{8}$	0.210	1310	1180	$1\frac{1}{16}$	0.715	4469	4022
$\frac{3}{4}$	0.324	2022	1820	$1\frac{1}{8}$	0.812	5075	4567
$\frac{7}{8}$	0.462	2885	2597	$1\frac{3}{16}$	0.915	5718	5145
$\frac{15}{16}$	0.540	3376	3038	$1\frac{1}{4}$	1.024	6403	5762

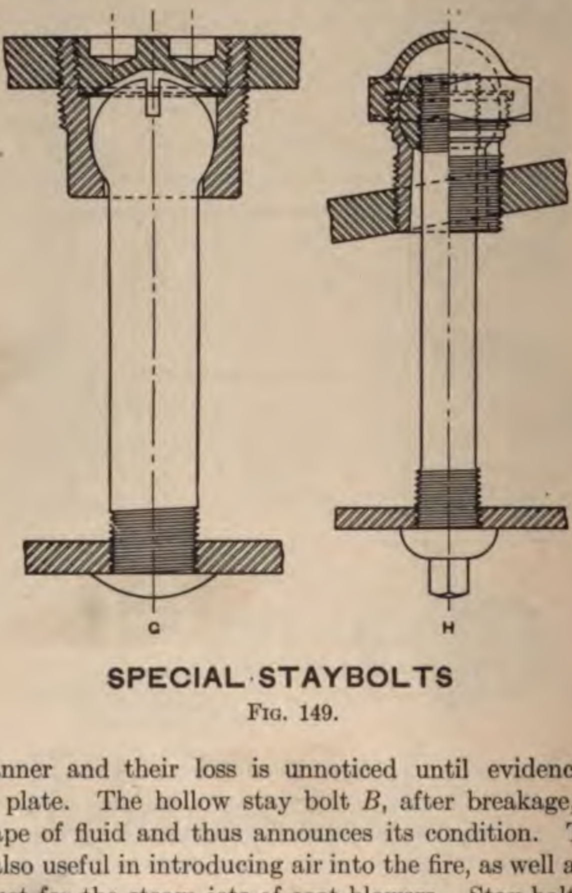
There are many different types of stay bolts a few of which are shown in Figs. 148 and 149. The chief difficulty in using stay bolts



TYPES OF STAYBOLTS

Fig. 148.

arises from the unequal expansion of the two sheets connected. One is usually in full contact with the fire while the other is heated to the temperature of the confined fluid only. There is, therefore, considerable relative motion between the two sheets. This conduces to the formation of cracks in the stay bolts near the inner surfaces of the sheets. Many stay bolts of the type *A* rupture in this manner and their loss is unnoticed until evidenced by a bulging plate. The hollow stay bolt *B*, after breakage, permits the escape of fluid and thus announces its condition. The axial hole is also useful in introducing air into the fire, as well as providing a duct for the steam jets of soot blowers. Stay bolt stock is usually procurable in long lengths, threaded continuously from end to end and necked down at intervals where it is desired to cut it off. One end is squared to facilitate screwing into place. Tapped holes for receiving stay bolts must have continuous thread through both sheets if the bolt is to screw to place easily and accurately.



SPECIAL STAYBOLTS

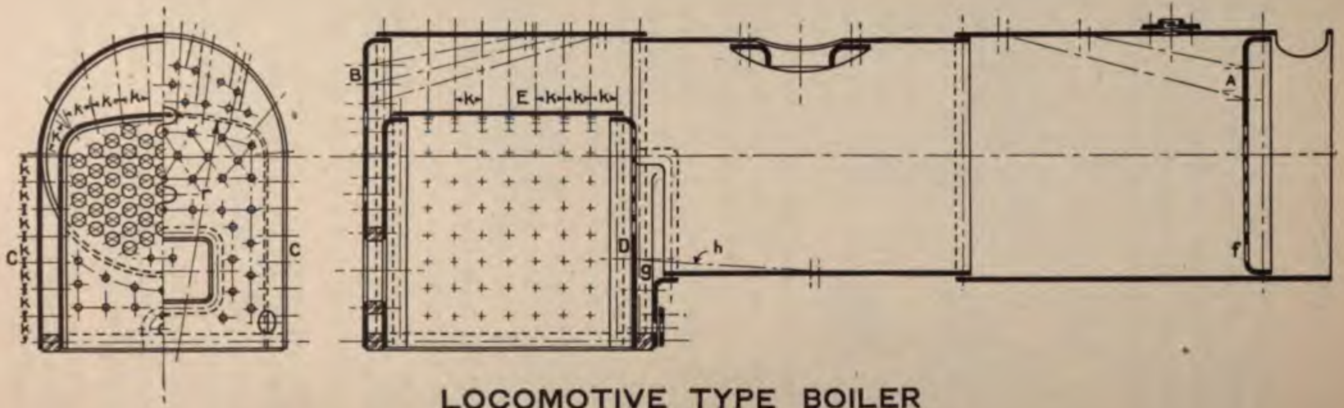
Fig. 149.

material is a very soft but reliable steel which will endure the ssarily severe usage. Some manufacturers still adhere to the of best grade wrought iron for the purpose. Having selected y bolt which represents a margin of safety over the condi- of the problem, the design of the staying may be considered plete.

hile discussing the subject of stay bolts it may be well to refer rious special forms adapted to severe conditions of contraction expansion. In Figs. 148 and 149, *CDEFG* and *H* illustrate us types adopted by the Pennsylvania Railroad and others se in locomotives. Flexibility is here insured by the spherical at one end of the bolt. In each case a sleeve has first to be ved into the boiler plate. The bolt itself is then put in place its end riveted over upon the surface of the furnace sheet. prevent leakage a soft copper washer is placed between the e and the cap nut, the latter being screwed down hard. In cases it is desirable to screw in the bolt from the fire box end, t copper washer similar to that described above being placed r its head. In this case the bolt enters a spherical nut which in the sleeve, and the cap nut and washer make the outer end . The outer end of these stay bolts need not always come ugh the plate perpendicularly but with extra long sleeves, an in- tion of 25 degrees as shown in *H* may be obtained. Frequently uble thickness of plate is used locally to give thread room for long sleeve. Supports and other external fixtures of various rs sometimes interfere with the external heads of stay bolts, hich case the form of flush-headed bolt shown in *G* should be . It is sometimes difficult to introduce the necessary stays out conflict. To avoid this, stay rods frequently have to be ged to two equivalent rods, and provided with spherical joints revent cramping.

aying in Locomotive Type Boilers. — The locomotive type of r, Fig. 150, presents many difficult problems in staying on unt of the construction of the fire box. There are five locali- which need attention in this respect:

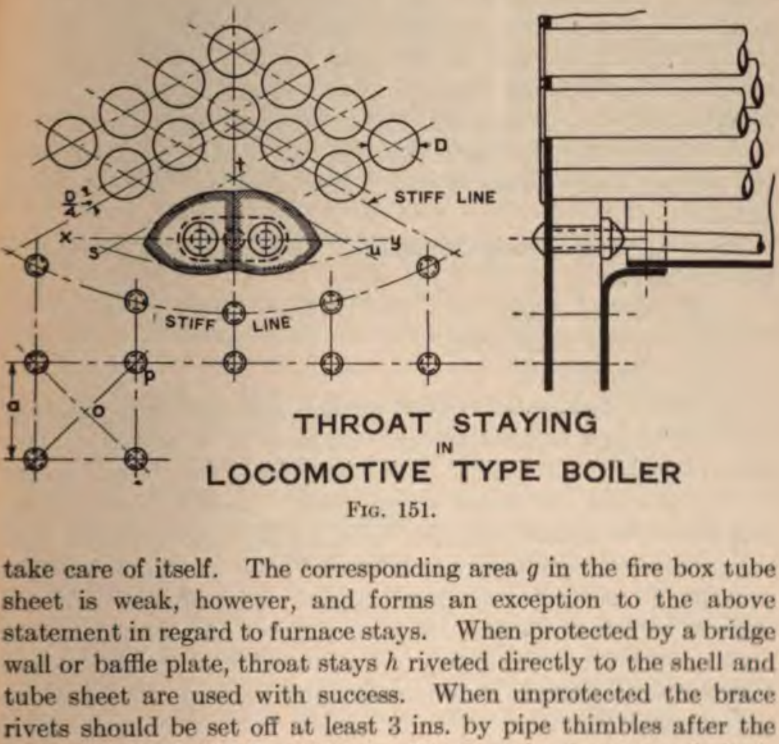
(*A*) *Smoke box tube sheet.*
(*B*) *Flat outer furnace wall.*
(*C*) *Water legs.*
(*D*) *Fire box tube sheet at throat.*
(*E*) *Furnace roof.*



LOCOMOTIVE TYPE BOILER
WITH
RADIAL STAYS

Fig. 150.

The smoke box tube sheet *A*, flat outer furnace wall *B* and the water legs *C* are stayed with diagonal braces and stay bolts in precisely the same manner as the corresponding portions of horizontal and vertical boilers. The fire box tube sheet *D* cannot generally be strengthened by stays of any type on account of the intensity of the heat. This is avoided at the top of the sheet by arranging the beaded tubes to approach uniformly near to the margin and thus to stiffen the sheet by their holding power. The location of the front handhole at *f* necessitates the omission of a group of tubes. The handhole cover is inherently stiff and will take care of itself. The corresponding area *g* in the fire box tube sheet is weak, however, and forms an exception to the above statement in regard to furnace stays. When protected by a bridge wall or baffle plate, throat stays *h* riveted directly to the shell and tube sheet are used with success. When unprotected the brace rivets should be set off at least 3 ins. by pipe thimbles after the manner described in Fig. 144, page 255, for tube sheet stays in horizontal boilers. This gives enough circulation around the stay rivets to prevent burning.



THROAT STAYING IN LOCOMOTIVE TYPE BOILER

FIG. 151.

Fig. 151 shows the method of placing and calculating throat stays. Stiff lines a distance $\frac{D}{4}$ back from the tube edges are drawn as heretofore. The circular row of stay bolts just below the throat forms the other line of rigid plate. From these stiff lines half the maximum plate spacing is set off, resulting in the unstayed area *stu*. The influence of the tubes reaches to *st* and *tu* and of the stay bolts to *su*. The unstayed portion is then divided by planimeter or otherwise into two equal areas above and below the horizontal line *xy*. One or more diagonal brace heads located symmetrically on this line will make the best possible provision for the flexible plate. The other ends of the braces are anchored well forward on the shell to avoid angularity. The shaded area shows the actual plate supported by the brace in this case, its boundaries being the locus of points equidistant from the rivet centers and stiff lines. If the distance from any point on the locus to the nearest stiff line is not more than half the diagonal *op* of a square having the maximum plate spacing *a* for its side, the arrangement may be considered safe. Care should be taken that the stayed area multiplied by the working boiler pressure does not overload the brace rivets or shank.

There are three general methods of staying the roof of locomotive fire boxes:

(*l*) *Radial stays.*
(*m*) *Slung crown bars.*
(*n*) *Crown bars.*

(*l*) *Radial stays.*— For the first method, Fig. 150, the crown sheet *E* of the furnace is made the arc of a circle of large radius *r*, for the greater part of its width. Where the roof joins the side sheets a curve of much less radius is employed. The spacing *kk* for the side sheets, as determined from Table XXXIX, is continued over the crown sheet, no additional stiffness being assumed for the arched plate in the roof. From the roof to the outer shell stay bolts are inserted with their axes as nearly as possible radial to the curve formed by the fire box sheet. The external course subjected to severe internal pressure forms a secure anchorage for the stay bolts thus located.

Having stepped off the circumferential pitch along the crown sheet the longitudinal spacing should next be given attention. Generally the pitch here used, *k*, is the same as that previously employed. Starting at the vertical side seam on the outside of the water leg at the rear, the pitch is stepped off, as shown, to a similar point at the front seam of the water leg. The stay bolts, therefore, fall on squares and the area of plate supported by each may be readily calculated. Care must be taken that conflict does not take place between the above radial stays and the diagonal braces employed on the flat exterior furnace sheet. The diagonal braces can sometimes be shifted slightly from a true radial position to avoid this difficulty.

Naturally the anchorage of the outer ends of radial stays to the shell takes place at an irregular angle. Unless very much out of a radial position this does not present special difficulty, as was explained in relation to the Tate stay bolt shown in Fig. 149*H*. This form or its equivalent should be used when the angle is of any considerable amount.

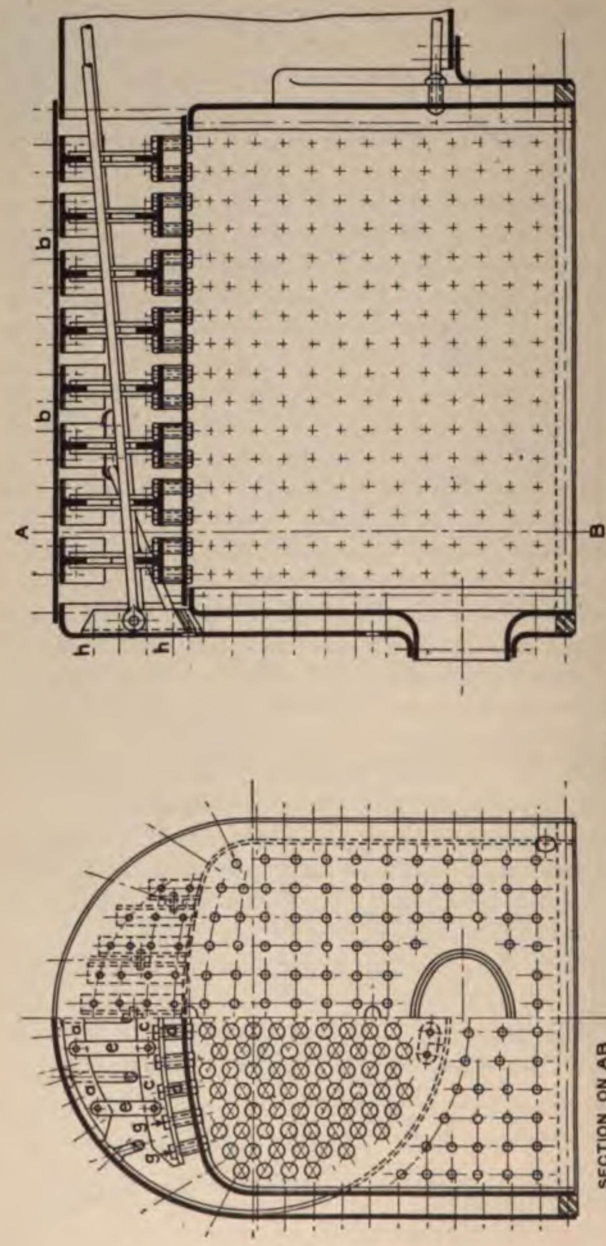
When radial stays are used under a dome, the anchorage in the outer shell is interrupted and some other form of stay must be used. Special forged lugs riveted to the inside of the dome and pinned to the stay bolts afford the usual means of fastening. This case generally occurs in the style of "wagon top" locomotive boilers in vogue until recently upon many railroads. The dome in this type is placed directly over the furnace on the rear sheet.

(*m*) *Slung crown bars.* — Fig. 152 shows a typical installation of slung crown bars. A series of structural steel T-beams *aa* curved to fit the exterior shell are firmly riveted at the necessary intervals to the sheet *bb* over the furnace. A corresponding set of T-beams *cc* are shaped to fit the furnace roof and held securely in place by thimbled bolts *dd*. The two T-beams are linked together by double ties *ee* at frequent intervals, the whole constituting a symmetrical system of supported points. With regular spacing the tension in each link may be taken as its proportionate share of the total load on the unstayed plate. The furnace roof beam is approximately a curved continuous girder of equal spans uniformly loaded. Neglecting the curvature its calculation is not difficult. If w is the uniformly distributed load per unit of length and L is the length of each span, the maximum bending moment, occurring over a support, for a continuous girder of indefinite length is

$$M = -\frac{wL^2}{12}, \qquad \qquad (131)$$

and similarly the bending moment at the center of each span is

$$M = +\frac{wL^2}{24}. \qquad \qquad (132)$$



SLUNG CROWN BARS IN LOCOMOTIVE TYPE BOILER

Fig. 152.

Very frequently these bending moments are exceeded by those caused by the overhung loads *gg* at the ends of the girder. In any event the T-beams should embody large factors of safety to resist the effects of corrosion. It must be borne in mind that such a calculation as the above is only approximate, the position of the girder with reference to its supports having much to do with the truth of the above equations.

The links and pins should be calculated for tension and bending respectively. The roof bolts are spaced precisely like the stay bolts previously described. Copper washers are used under the bolt heads to keep them tight and the thimbles not only permit circulation but enable the whole system to be assembled under a good degree of initial tension. Slung crown bars are adaptable as well to flat surfaces over furnaces and combustion spaces. When the dome or steam outlets prevent the use of anchor beams, the links are pinned to specially forged lugs riveted directly to the sides of the dome or shell. With this form of staying there is generally considerable difficulty in arranging diagonal braces to take care of the flat external furnace sheet *hh*. Stiff angles riveted to the flat plate and anchored well forward to the shell by one or more eye-braces form the best solution. An occasional diagonal brace may be used where room permits.

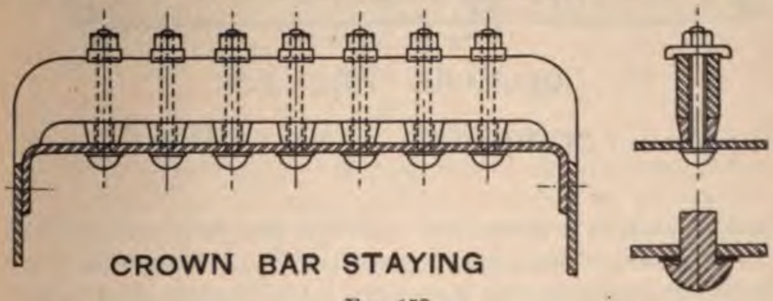
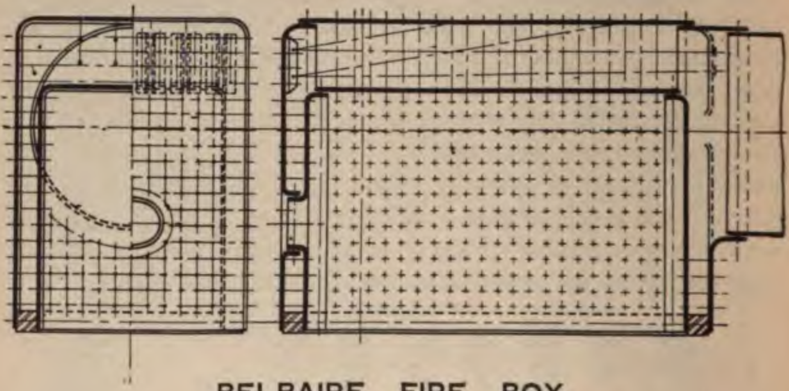


Fig. 153.

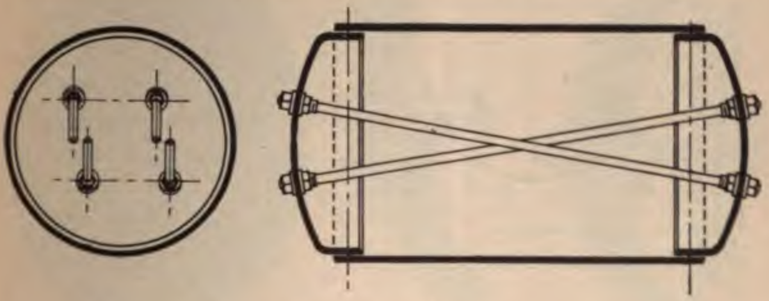
(*n*) *Crown bars.* — Crown bars on account of their unreliability are rapidly becoming obsolete. They are best adapted to the flat tops of furnaces and combustion chambers. Fig. 153 shows a typical arrangement of crown bars. The flat sheet is crossed at necessary intervals by straight beams which receive their support at the flange forming the boundary of the furnace. The bars are made of wrought or cast iron and are either bored or made double to permit the bolts to pass down through to the crown sheet. If double, a clip washer unites each pair and a tapered thimble permits the bolts to be assembled in tension. Naturally the support at the flange is very insecure, and, receiving as it does at each end half the entire roof load, local distortion is liable to occur. The feet frequently spread and slip off the flange so that the holding power of the entire beam is lost. The crown bars are simple beams supported at the ends and loaded symmetrically at a system of points. Their calculation is consequently very simple. This form of staying is rarely used at present except upon the roofs of the combustion chambers of marine boilers. The spacing of the bolts follows the principles set down in the previous cases of plate staying.



BELPAIRE FIRE BOX
IN
LOCOMOTIVE TYPE BOILER
FIG. 154.

In order to simplify as far as possible the problem of staying locomotive boilers, a fire box, designated by the term "Belpaire" from the name of the inventor, has been widely used. The principle embodied consists in the use of flat sheets and direct through stays wherever possible. Fig. 154 illustrates the general arrangement. The vertical and transverse stays are of the direct flexible type with special seats and cap nuts at all cool ends. The longitudinal braces consist of diagonal stays fastened to a shell lug at one end and pinned between heavy angles at the head. The angles are vertical and riveted symmetrically to the external flat furnace sheet. A system of throat stays takes care of the flexibility in that locality. To avoid conflict among the stays, an occasional one is either slightly displaced from its true position or bridging is employed. Except for the expense this is an excellent type of construction.

When subjected to severe pressures the inherent stiffness of dished heads is not sufficient to resist distortion and stays must be provided for the purpose. Stiffeners such as tees or angles may be shaped to fit the head and riveted securely to it. Fig. 155 shows a method employed in steam drums where through rods set at an angle bind the heads together. The rods are inserted in a position approximately normal to the curvature of the heads and the nuts are made tight with thin copper gaskets. When several pairs of rods are required it is necessary to make a special spotting in the head in order to provide a suitable seat for the nuts. This can be done at the time of manufacture with but little difficulty.



THROUGH STAYS
FOR
DISHED HEADS
FIG. 155.

69. Boiler Mountings. — *Manholes.* — In most boilers it is desirable, either from necessities arising in the manufacture or from the subsequent requirements of cleaning and inspection, that provision be made to enable a man to enter the shell. In steam drums having dished heads, both of which are externally convex, it is always necessary to provide entrance for driving the rivets. In large cylindrical boilers the same necessity presents itself in the repair and maintenance of the staying system. The standard size of manholes is 11 by 15 ins. In order to weaken the shell as little as possible they should be placed with their narrow dimension lengthwise of the cylinder to which they are attached. A heavy internal ring double-riveted to the shell is designed to compensate for the loss of strength due to cutting the hole. Fig. 156 shows the general appearance and dimensions of a standard forged steel manhole ring with seam. This type is used where pressures do not exceed 150 lbs. per sq. in. In order to displace as little of the steam space as possible the manhole frame is kept shallow. The gasket is generally one inch wide and liberal fillets are used in pressing its seat from the surrounding frame. The special double-riveted seam shown is the one recommended by the Massachusetts Boiler Rules. Such seams should always be struck on the flat plate before rolling.

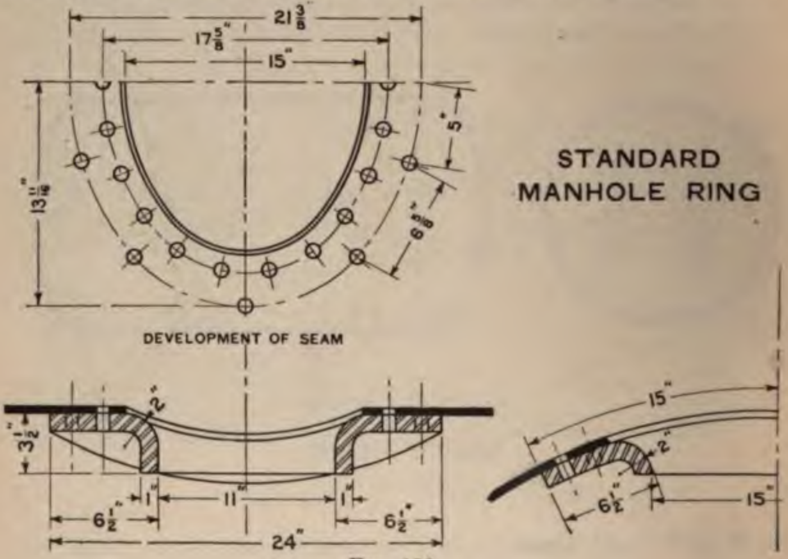


FIG. 156.

Manhole covers were formerly made of cast iron, but the use of this material has been largely superseded by steel castings or pressed steel. In cast covers the manhole bolt passes through a hole and is riveted to place on the inside. This arrangement is a source of weakness as well as leakage. Corrugated pressed steel covers, as illustrated in Fig. 157,* are at present widely employed. The bolt is anchored to a plate held in place by the corrugations.

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* The Lukens Iron Works.

Yokes are also made of pressed steel, the form being that shown in the illustration. Foot room must be provided for the yoke in the region of the seam.

In Scotch boilers manholes are often located in the front head just above or below a pair of furnace tubes. In such cases the tube sheet must be heavily reinforced by a pad securely riveted to it. Figs. 158 and 159 show the respective arrangement of such manholes. In Fig. 158 the anchorage of a stay rod from the rear tube sheet is incorporated in the reinforcement of the manhole opening. A heavy dished cover is used to close the manhole. The rivets must be smoothly countersunk on the inside ends to provide a flat seat for the gasket.

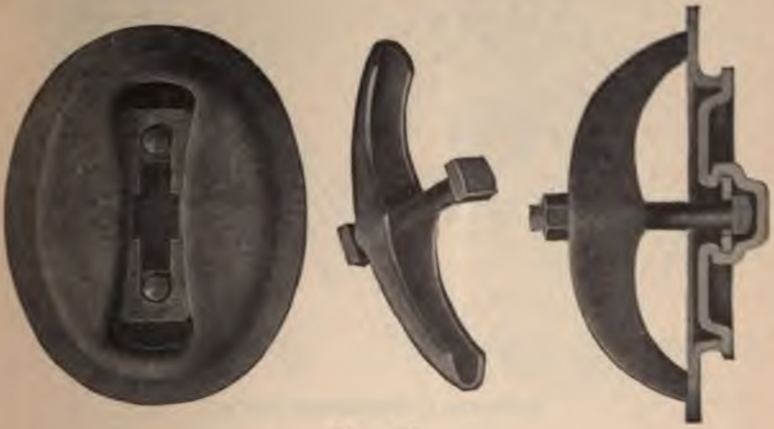
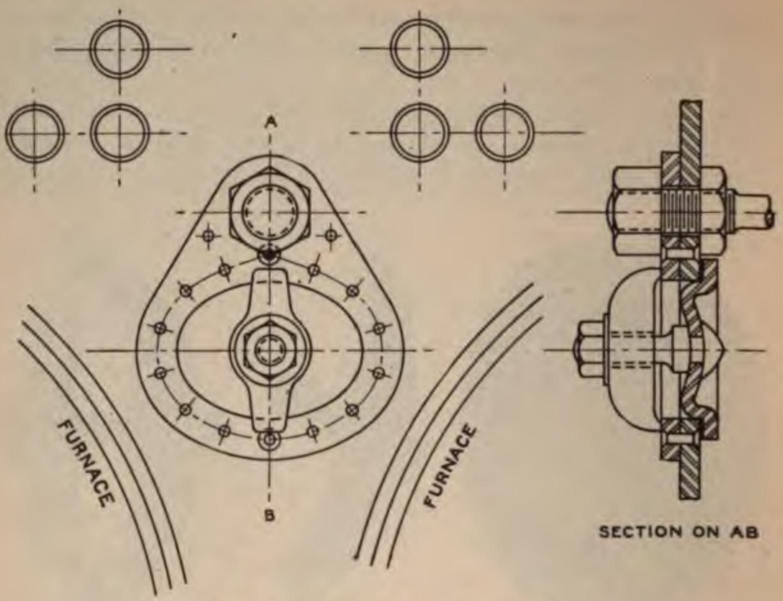


Fig. 157.

In Fig. 159 three stay rod ends are seated upon the reinforcement. Since these mountings are well removed from the fire there is no special need of avoiding large masses of metal and their dimensions may be as ample as the strength requires.

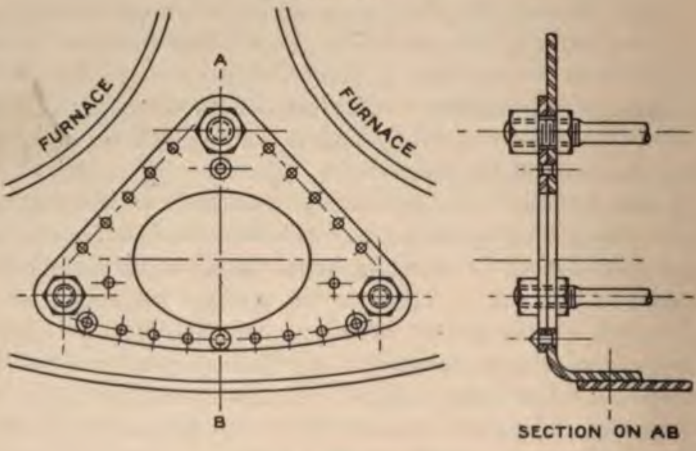
Handholes. — To permit access to the sheets of a boiler for cleaning and repair, handholes of elliptical pattern should be installed at all necessary points. The plate over the fire is especially liable to injury if accumulations of sediment are allowed to collect. All crown sheets should, therefore, be easily reached from without. Water legs are natural traps for mud and, unless the water circulation is maintained, will clog up causing burned plate. In locomotives a perforated wash out pipe is sometimes



MANHOLE REINFORCEMENT

ABOVE FURNACES

FIG. 158.



MANHOLE REINFORCEMENT

BELOW FURNACES

FIG. 159.

permanently installed just above the mud-ring for cleaning purposes. At all events corner handholes of the type shown in Fig. 160 should be provided so that there shall be complete access to all portions of the water legs, as well as to the space above the fire door. The standard handhole, yoke and cover for use on flat sheets is shown in Fig. 161. Cast iron was formerly used for the cover and yoke but with increasing pressures, steel castings or forgings have been largely adopted. In locating handholes room must be secured for the necessary seating of the gasket. The latter is generally $\frac{3}{4}$ in. wide by $\frac{1}{8}$ in. thick and is made with special reference to durability in the presence of heat and corrosion. The yoke or crab is generally shaped as shown in the figure. Four-footed crabs have been used to secure a better joint near the ends of the handhole opening. A lip should always be cast around the cover to assist in its seating so that the full gasket width shall be available in making the joint tight. Handhole covers are necessarily so thick that they should be removed as far as possible from contact with the fire. When placed in the region of intense heat, thin dished steel covers, convex against the pressure, have been used satisfactorily. The threaded end of the handhole bolt should not project beyond the nut since burning and corrosion will injure the thread to such an extent that the nut cannot be removed when desired.

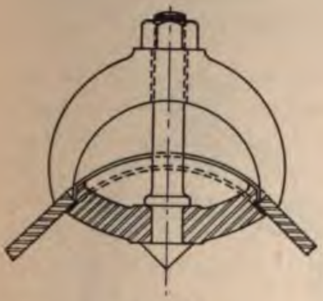


FIG. 160.

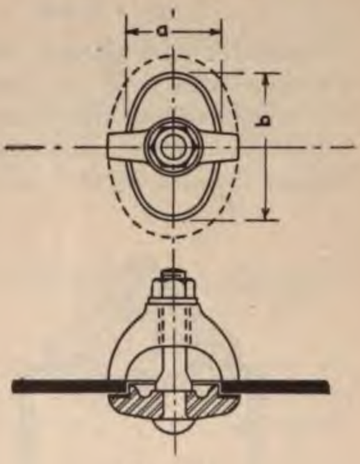
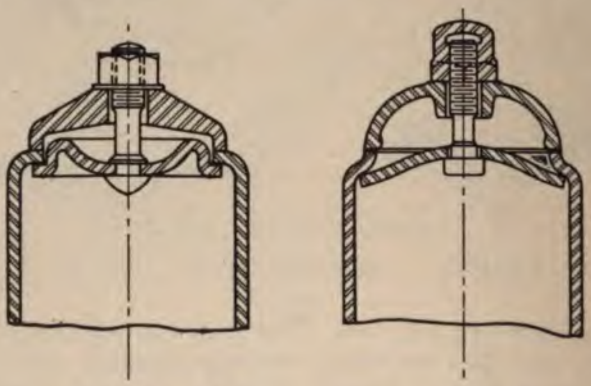


FIG. 161.

Openings of this class are shaped as true ellipses with their minor and major axes in the approximate ratio of two to three. The standard sizes usually found are as follows:

2¼ ins. x 3¼ ins.
2⅝ ins. x 3¾ ins.
3 ins. x 4½ ins.
3½ ins. x 5 ins.
4 ins. x 6 ins.
5½ ins. x 8 ins.

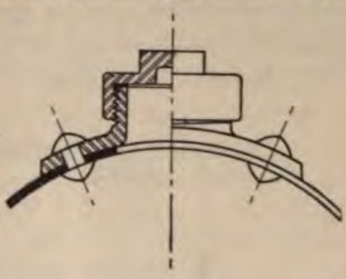
Handholes must generally be located opposite each of the tubes where the latter enter the headers of water tube boilers to facilitate expanding the tubes to place. Such handholes must be provided with elliptical milled seats if the covers are to be inserted through the opening and an external yoke used. To avoid the necessity of machining an ellipse, circular handholes with internal yokes have been designed. The cover is wholly external and rests on a ground seat. The yoke and bolt bind it to place as shown at the right of Fig. 162. Leakage at the nut seat may be prevented by the use of a copper washer. To enable the bolt to adjust itself and insure a fair bearing for the cover, the bolt head is sometimes given a spherical upper surface.



OUTSIDE YOKE INSIDE YOKE

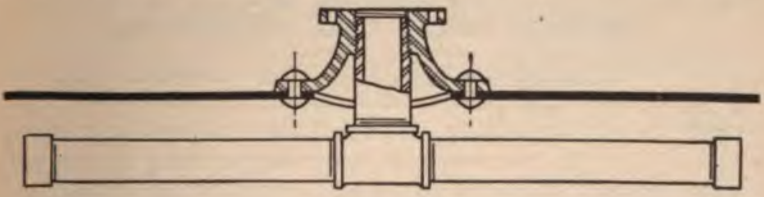
HANDHOLE COVERS
FIG. 162.

Washout Plugs. — Where there is not room for a handhole of any of the above sizes a smaller opening, known as a washout, is used. Fig. 163 shows the usual proportions. This detail consists of a pad riveted externally to the shell, the threaded opening being covered with a cap and copper gasket. The pad is forged to the curvature of the plate against which it is to fit. For the introduction of hose and scraping rods, these openings are of much value in the restricted portions of steam generators. The threaded nozzle of the washout is made to conform to one of the smaller pipe sizes so that if necessary the cap may be replaced temporarily with an ordinary pipe cap.



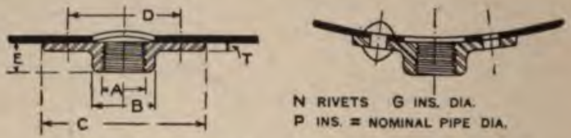
CORNER WASHOUT
FIG. 163.

Dry Pipes. — When there is not room for external dome or drum, or when the pressure in the boiler forbids the use of such attachments, a dry pipe is inserted in the ordinary steam space. Fig. 164 shows an inexpensive method of building up and attaching a dry pipe with standard pipe fittings. There is generally stock enough in the nozzle to permit the hole to be tapped for the reception of a pipe nipple. A tee with capped branches completes this form of dry pipe. The upper semi-circumference of each branch is usually perforated with holes $\frac{1}{8}$ in. in diameter through which the steam is drawn. By this means the splash of ebullition is prevented from entering the steam pipe. In locomotive boilers a perforated dry pipe is slung from the ring seams and leaves the boiler through the upper portion of the front tube sheet.



DRY PIPE CONNECTION
UPPER HALF PERFORATED
FIG. 164.

Forged Pipe Flanges. — To avoid the brittleness of cast metal, especially when subjected to the pressures employed in driving rivets, forged pipe pads have been placed upon the market. They consist of a simple disc of metal from the center of which is forged up a pipe hub. The latter contains sufficient depth to give the requisite number of perfect threads for the pipe in question. The hole in the boiler plate to which the flange is attached need not, therefore, be threaded. Such flanges are usually attached to the boiler by ⅞ in. rivets a single row being generally sufficient for pressures up to 150 lbs. per sq. in. Fig. 165 shows the usual proportions of these attachments and Table XLVII gives the corresponding dimensions. Such flanges have to be curved to fit the circumference of the shell to which they are riveted.



FORGED STEEL PIPE FLANGE

Fig. 165.

TABLE XLVII.

PROPORTIONS OF STANDARD FORGED STEEL PIPE FLANGES.

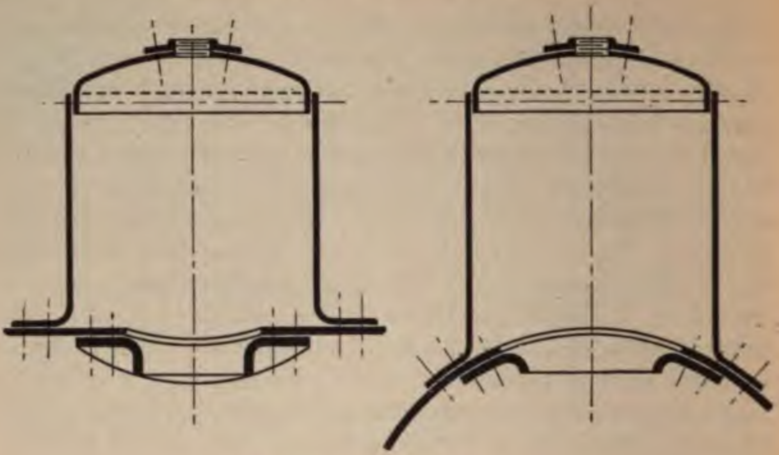
Nominal diam. of pipe. *P*.	For significance of letters see Fig. 165, p. 282. All dimensions in inches.							
	C	*E*	*T*	*A*	*B*	*D*	*N*	*G*
Ins.								
¾	6	1	5/16	1.05	1¾	3¾	6	⅞
1	6	1	5/16	1.32	2	4	6	⅞
1¼	6½	1	5/16	1.66	2¼	4¼	6	⅞
1½	7	1¼	⅜	1.90	2⅝	4⅝	6	⅞
2	8	1½	⅜	2.38	3⅛	5⅛	8	⅞
2½	8½	1½	⅜	2.88	3⅝	5⅝	8	⅞
3	9	1½	⅜	3.50	4¼	6¼	8	⅞
3½	9½	1½	7/16	4.00	4⅞	6⅞	10	⅞
4	10	2	7/16	4.50	5⅜	7⅜	10	⅞
4½	10½	2	½	5.00	6	8	12	15/16
5	11½	2	½	5.56	6⅝	8⅝	12	15/16
6	12½	2	½	6.63	7⅝	9⅝	14	15/16
7	14	2½	⅝	7.63	8⅞	10⅝	16	15/16
8	15	2½	⅝	8.63	9⅞	11⅝	16	15/16

Domes. — When the steam space of the boiler is necessarily restricted, additional volume may be obtained by attaching a steam dome or drum. Such members are generally cylindrical in shape with dished heads. The axis of the cylinder may be either vertical or horizontal, depending upon the head room obtainable above the boiler. Horizontal steam drums are usually attached directly to the boiler shell by special forged flanges riveted to place. The use of bolted cast iron flanges for this purpose has been largely discontinued. The dished heads in such drums are frequently stayed as illustrated in Fig. 155 on page 275. In order to assemble both of the heads with convex contour outward, an outlet must be provided in the shell through which a holder-on may be introduced to support the rivets while being driven from without. Manholes are frequently located in the dished heads for this purpose.

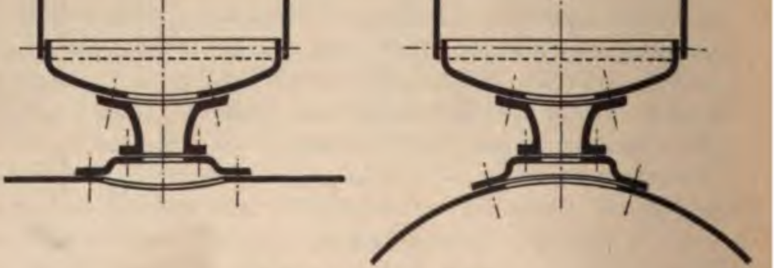
Domes of the vertical type are either riveted directly to the shell or bolted to it by pipe flanges. Fig. 166 shows one method by which a dome is connected directly to the cylindrical shell. As a precaution against rupture a standard manhole ring is first riveted to the course as if for that use alone. The dome, flanged at its lower end, is then double-riveted to the shell. This method provides sufficient opening for the exit of steam and insures the safety of the boiler. Sometimes a heavy saddle ring is fitted to the outside of the boiler and the dome flange riveted through it to the shell. The strength of such members is a disputed point and many accidents have happened due to the removal of excessive amounts of plate for such connections.

A second method of attaching a vertical steam drum is shown in Fig. 167. Two nozzles, one of high and one of low pattern, are arranged as shown in the figure, each being riveted to the proper member. When bolted together with a suitable gasket this provides a durable and satisfactory steam drum connection.

Fire Door Openings. — Two arrangements of the door opening through the water legs of locomotive and vertical boilers are shown in Fig. 168. In small boilers a wrought iron or cast steel frame may be used around door openings as shown at the left. The corners of the opening should be rounded to a liberal radius and the rivets driven in the same manner as at the mud-ring. It is generally impossible to drive such rivets by machine on account of inaccessibility. The chief difficulty with this type of door frame



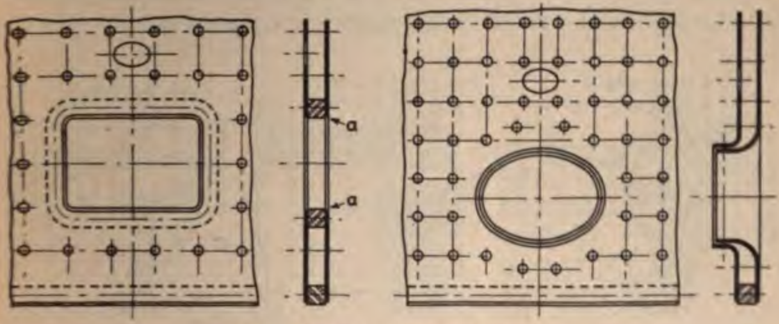
DOME ATTACHMENT
FOR
LOCOMOTIVE TYPE BOILER
Fig. 166.



ATTACHMENT
OF
VERTICAL STEAM DRUM
Fig. 167.

is the liability of burning at *a* where the thickness of metal prevents the circulation from keeping the plate cool.

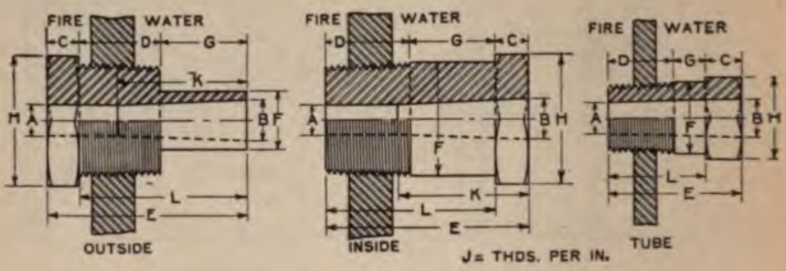
To obviate the above difficulty and also to provide for the necessities of the riveting machine, the type of opening shown in Fig. 168 at the right is most widely used at present. By the careful use of suitable machinery both the inner and outer sheets may be flanged without injury to a common joint having its pitch line at least $1\frac{3}{4}$ ins. outside of the adjacent plate. This expedient maintains a water circulation around the door and provides sufficient clearance for the use of the riveting machine. The outer plate is commonly allowed to project slightly beyond the inner so as to form a ledge against which the calking tool may rest. As was remarked above handholes must always be provided above door openings in order to make possible the removal of the sediment which would naturally accumulate at that point.



FIRE DOOR OPENINGS
IN
WATER LEGS

Fig. 168.

Fusible Plugs. — In every steam generator there are portions especially susceptible to injury from overheating in the event of low water. Such portions should be protected from extensive injury by the use of a hollow plug filled with a metal fusing at a comparatively low temperature. Fig. 169 shows three types of plugs recently recommended by the Massachusetts Board of Boiler Rules. The required locations of these safety devices is accurately specified in the Code of Rules summarized in Chap. I, page 36. When the form of the steam generator necessitates the insertion of the plug from the fire side of the sheet the latter is designated as an outside plug. When access to the desired locality can only be obtained from the inner or water side of the sheet, the type is designated as an inside plug. A third style screwed into the tubes of vertical boilers is shown at the right. These safety devices are required by law to be filled with pure Banca tin, the melting point of which is about 445° F. This figure represents a safe margin of



PROPORTIONS OF FUSIBLE PLUGS

Fig. 169.

temperature above that of saturated steam at ordinary pressures. Fusible plugs are not ordinarily allowed to come in contact with highly superheated steam. The efficacy of these devices in preventing accidents can only be assured by keeping the fusible metal clean where it comes in contact with the water. It is generally advisable to renew fusible plugs annually. The recommended proportions for fusible plugs are given in Table XLVIII. All steam generators should be protected by their use where there is the least probability of danger from low water.

Feed Pipes. — The feed water should enter the boiler in such a manner as to mingle with the contents at the coolest portion and to assist the natural circulation. This can best be accomplished in horizontal boilers by tapping the front head for a feed pipe bushing just above the level of the tubes. With girder stays the bushing may pass through the center of the channel. In order to withstand corrosion the bushing is usually made of brass and corresponds in size to that of a suitable pipe tap. The threads within taper from opposite directions, the bushing thus forming a coupling for the internal and external feed pipes. Brass pipe is

TABLE XLVIII.

PROPORTIONS FOR STANDARD FUSIBLE PLUGS.

Recommended by Massachusetts Board of Boiler Rules.

Outside Pattern.

For pressures less than 150 lbs. per sq. in.

Nominal size of pipe tap.	All dimensions in inches.										
	A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*L*
Ins.											
1/2	1/2	19/32	3/8	3/4	2 1/8	11/16	1	15/16	14	1 1/2	1 3/4
3/4	1/2	19/32	3/8	7/8	2 1/4	3/4	1	1 3/16	14	1 1/2	1 7/8
1	1/2	19/32	3/8	15/16	2 5/16	25/32	1	1 3/8	11 1/2	1 1/2	1 15/16
1 1/4	1/2	19/32	3/8	1 3/16	2 9/16	13/16	1	1 13/16	11 1/2	1 1/2	2 3/16
1 1/2	1/2	19/32	7/16	1 5/16	2 3/4	7/8	1	2 1/8	11 1/2	1 1/2	2 5/16

For pressures of 150 lbs. per sq. in. and above.

1/2	3/8	15/32	3/8	3/4	2 1/8	9/16	1	15/16	14	1 1/2	1 3/4
3/4	3/8	15/32	3/8	7/8	2 1/4	5/8	1	1 3/16	14	1 1/2	1 7/8
1	3/8	15/32	3/8	15/16	2 5/16	21/32	1	1 3/8	11 1/2	1 1/2	1 15/16
1 1/4	3/8	15/32	3/8	1 3/16	2 9/16	11/16	1	1 13/16	11 1/2	1 1/2	2 3/16
1 1/2	3/8	15/32	7/16	1 5/16	2 3/4	3/4	1	2 1/8	11 1/2	1 1/2	2 5/16

Inside Pattern.

For pressures less than 150 lbs. per sq. in.

1/2	1/2	19/32	3/8	3/4	2 1/8	27/32	1	15/16	14	1 1/2	1 3/4
3/4	1/2	19/32	3/8	7/8	2 1/4	1 1/16	1	1 3/16	14	1 1/2	1 7/8
1	1/2	19/32	3/8	15/16	2 5/16	1 5/16	1	1 3/8	11 1/2	1 1/2	1 15/16
1 1/4	1/2	19/32	3/8	1 3/16	2 9/16	1 21/32	1	1 13/16	11 1/2	1 1/2	2 3/16
1 1/2	1/2	19/32	7/16	1 5/16	2 3/4	1 29/32	1	2 1/8	11 1/2	1 1/2	2 5/16

For pressures of 150 lbs. per sq. in. and above.

1/2	3/8	15/32	3/8	3/4	2 1/8	27/32	1	15/16	14	1 1/2	1 3/4
3/4	3/8	15/32	3/8	7/8	2 1/4	1 1/16	1	1 3/16	14	1 1/2	1 7/8
1	3/8	15/32	3/8	15/16	2 5/16	1 5/16	1	1 3/8	11 1/2	1 1/2	1 15/16
1 1/4	3/8	15/32	3/8	1 3/16	2 9/16	1 21/32	1	1 13/16	11 1/2	1 1/2	2 3/16
1 1/2	3/8	15/32	7/16	1 5/16	2 3/4	1 29/32	1	2 1/8	11 1/2	1 1/2	2 5/16

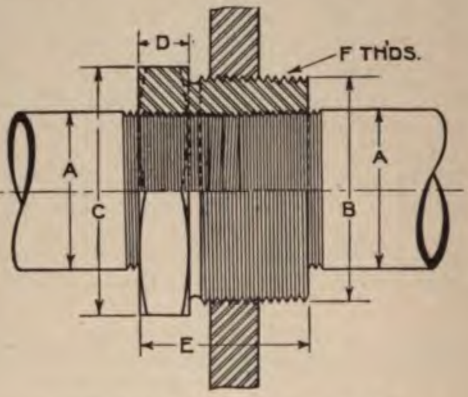
Tube Plug.

For all pressures.

3/8	5/16	3/8	3/8	3/4	1 1/2	11/16	3/8	13/16	18	....	1 3/8
1/2	3/8	7/16	3/8	3/4	1 1/2	27/32	3/8	13/16	14	....	1 3/8

generally used for the purpose throughout. The internal feed pipe runs about two-thirds the length of the boiler, crosses at right angles and is arranged with elbow and nipple to discharge the water into the downward circulation near the shell. Such feed pipes are generally supported by small brass stirrups attached to adjacent stay rods.

In vertical boilers a forged steel pipe flange riveted to the outside is arranged to receive the feed pipe bushing. The internal feed pipe enters one of the avenues in the tube arrangement and discharges the water near the center of the boiler.



FEED PIPE BUSHING
Fig. 170.

The feed pipes of Scotch and locomotive boilers generally enter through the side and discharge the water near the shell as in the case of the horizontal boiler.

Fig. 170 illustrates the standard feed pipe bushing and Table XLIX gives the usual dimensions embodied in its construction.

Nozzles. — The steam exits from boilers are usually made from either forged or cast steel. A flaring body with heavy ring at the bottom provides not only for the quiet withdrawal of steam from the boiler but for a strong riveted seam at the base as well. In order to rivet the seam by machine the center line of the plunger must have at least $1\frac{3}{4}$ ins. clearance beyond the lip of the pipe flange. The most critical point in the design of the nozzle is at the juncture of the body with the top flange. Unless good material and liberal fillets are employed at this point fracture is liable to occur. The top flange is designed to meet the requirements of standard pipe flanges.

TABLE XLIX.

PROPORTIONS FOR STANDARD FEED PIPE BUSHINGS.

Nominal size of feed pipe.	For significance see Fig. 170, p. 288. All dimensions in inches.						
	A	*B*	*C*	*D*	*E*	*F*	*G*
Ins.							
¾	1.050	1.660	1¾	⅝	1¼	11½	14
1	1.315	1.900	2	¾	1½	11½	11½
1¼	1.660	2.375	2½	¾	1¾	11½	11½
1½	1.900	2.375	2½	¾	1¾	11½	11½
2	2.375	2.875	3	⅞	2¼	8	11½
2½	2.875	3.500	3¾	1	2¾	8	8

Two types of flange, the high and the low, are found on the market. The high type, Fig. 171, permits the use of through bolts. In case of rupture the latter may be easily renewed. When head room is at a premium the low type of nozzle must be

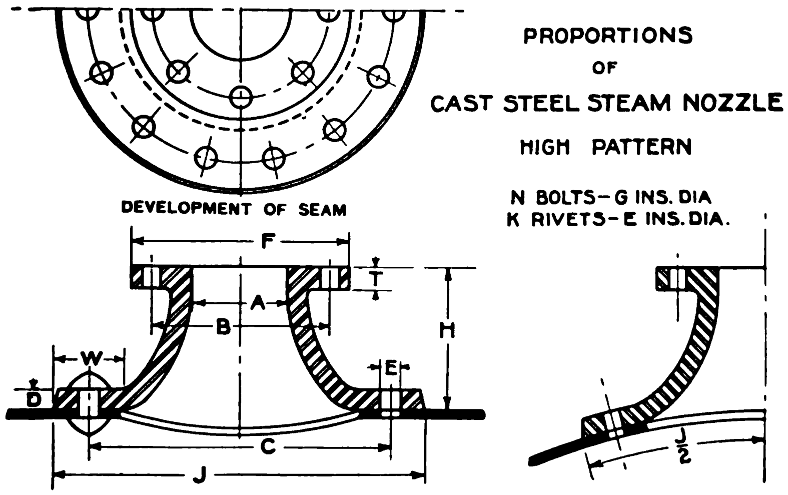
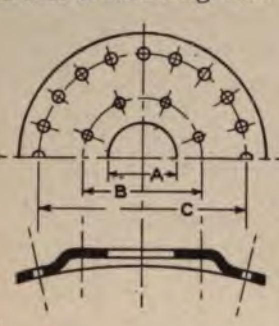


Fig. 171.

used as illustrated in Fig. 172. Stud bolts are never a desirable means of fastening from the standpoint of breakage and as well as leakage. The channel through which the steam is withdrawn in this type of nozzle is also very abrupt.



STEAM NOZZLE
LOW PATTERN
FIG. 172.

Table L gives the standard dimensions of the Taylor seamless forged steel boiler nozzle as manufactured by the American Spiral Pipe Works. The proportions of the low pattern, Fig. 172, also conform approximately to Table L as far as the letters signify.

The riveting at the base of the nozzle has not been absolutely standardized and may be varied at the will of the designer. The proportions given in Table L are suitable ones for medium pressures and average service. For severe conditions there is room enough in the base ring to permit the use of a double-riveted seam.

TABLE L.

PROPORTIONS OF FORGED STEEL BOILER NOZZLES.

American Spiral Pipe Works.

Nominal diam. of pipe. *A*	Upper flange dimensions.					Approx. height. *H*	Lower flange dimensions.					
	Diam. *F*	Thickness. *T*	Bolt circle diam. *B*	No. of bolts. *N*	Diam. of bolts. *G*		Diam. *J*	Thickness. *D*	Width. *W*	No. of rivets. *K*	Diam. of rivets. *E*	Diam. of rivet circle. *C*
Ins.	Ins.	Ins.	Ins.		Ins.	Ins.	Ins.	Ins.	Ins.		Ins.	Ins.
1½	6	¾	4⅝	4	¾	5	9	11/16	3	10	⅞	6
2	6½	⅞	5	4	¾	5	9	11/16	3	10	⅞	6
2½	7½	1	5⅞	4	⅞	5	12½	9/16	3¼	12	⅞	9¼
3	8¼	1	6⅝	8	¾	5	13	9/16	3¼	12	⅞	9¾
3½	9	1⅛	7¼	8	¾	6	14	⅝	3½	14	⅞	10½
4	10	1⅛	7⅞	8	⅞	6	15	⅝	3½	14	⅞	11¼
4½	10½	1¼	8½	8	⅞	6	15½	⅝	3¾	16	⅞	11½
5	11	1¼	9¼	8	⅞	6	16	⅝	4	16	15/16	12
6	12½	1¼	10⅝	12	⅞	7	17½	11/16	4¼	18	15/16	13¼
7	14	1 5/16	11⅞	12	1	7	19	11/16	4½	18	1	14½
8	15	1⅜	13	12	1	7	20	¾	4¾	20	1	15¼

Pipe Flanges. — In 1894 the A.S.M.E. proposed the standardized dimensions given in Fig. 173 and Table LI for cast iron pipe flanges. These sizes have remained practically without change to the present day except in pressures above 100 lbs. per sq. in. In 1901 a Committee of Manufacturers standardized extra heavy steam pipe flanges as given in the lower part of Table LI.

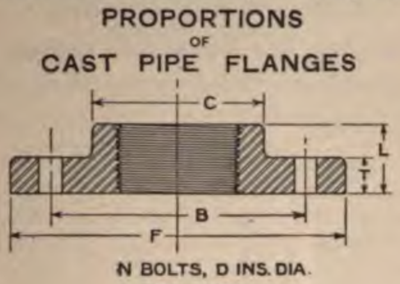


FIG. 173.

The bolt holes in pipe flanges are drilled $\frac{1}{8}$ in. larger in diameter than the bolts specified in Table LI to insure ease in adjustment.

Some manufacturers provide a third series of cast steel flanges made integral with the pipe or fitting and designed for 350 lbs. per sq. in. pressure and superheating temperatures up to 800° F.

In addition to the screwed fittings described above there are many special types of pipe flanges designed for severer service. These may be roughly divided into three classes: 1° Welded Flanges, 2° Shrunk and Peened Flanges, 3° Clamped Pipe Joints.

To insure against leakage under high pressures a forged steel pipe flange may be welded directly to the pipe. Or again the pipe may be partially threaded into the flange and the joint sealed with soft steel under the oxy-acetylene flame.

When the service is not so severe the flange ring may be shrunk upon the unthreaded pipe and the latter peened out into a flaring recess at the flange surface.

The third form of pipe union, known some years ago as the Van Stone joint and at present bearing a variety of names, consists of turning a 90° flange upon the ends of the pipes to be united. Heavy bolt rings inserted loosely upon the pipe draw the flanges tightly together and make the joint. One great advantage of this method consists of the possibility of turning the pipes relatively to one another through any desired angle without encountering the difficulty of unscrewing rusty pipe threads.

TABLE LI.

PROPORTIONS OF STANDARD PIPE FLANGES.

Cast Iron, Gun Iron or Steel Castings.

For pressures not over 100 lbs. per sq. in.

Nominal diam. of pipe.	All dimensions in inches.						
	F	*T*	*L*	*C*	*B*	*N*	*D*
Ins.							
2 1/2	7	11/16	1	3 5/8	5 1/2	4	5/8
3	7 1/2	3/4	1	4 5/16	6	4	5/8
3 1/2	8 1/2	13/16	1 1/16	4 7/8	7	4	5/8
4	9	13/16	1 1/8	5 3/8	7 1/2	8	3/4
4 1/2	9 1/4	15/16	1 1/4	5 13/16	7 3/4	8	3/4
5	10	15/16	1 1/4	6 7/16	8 1/2	8	3/4
6	11	1	1 3/8	7 9/16	9 1/2	8	3/4
7	12 1/2	1 1/16	1 1/2	8 5/8	10 3/4	8	3/4
8	13 1/2	1 1/8	1 5/8	9 11/16	11 3/4	8	3/4

For pressures from 100 to 250 lbs. per sq. in.

Nominal diam. of pipe.	*F*	*T*	*L*	*C*	*B*	*N*	*D*
2 1/2	7 1/2	1	1 5/16	4	5 7/8	4	3/4
3	8 1/4	1 1/8	1 5/8	4 5/8	6 5/8	8	5/8
3 1/2	9	1 3/16	1 3/4	5 1/4	7 1/4	8	5/8
4	10	1 1/4	1 13/16	5 3/4	7 7/8	8	3/4
4 1/2	10 1/2	1 5/16	1 7/8	6 3/16	8 1/2	8	3/4
5	11	1 3/8	2	6 3/4	9 1/4	8	3/4
6	12 1/2	1 7/16	2	7 15/16	10 5/8	12	3/4
7	14	1 1/2	2 3/16	9	11 7/8	12	7/8
8	15	1 5/8	2 1/4	10 5/8	13	12	7/8

Pipe-Sizes. The manufacture of steam and water pipe consists of butt- or lap-welding together strips of wrought iron or steel plate so shaped as to form the correct diameter of the tube. The flat plate with edge scarfed is known as "skelp" and must be of exactly the right width to give the size of pipe desired when rolled up and welded.

In addition to the above there are many forms of seamless tubing drawn both hot and cold.

The standard dimensions of steam and water pipes as taken from the National Tube Co.'s Book of Standards are given in Table LII.

TABLE LII.

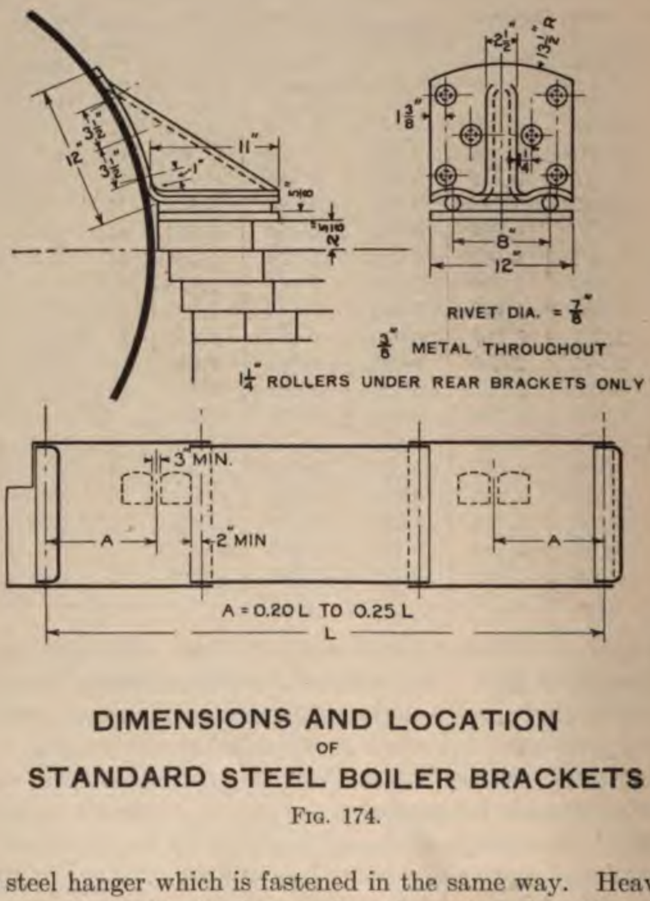
PROPORTIONS OF STANDARD WROUGHT IRON PIPE.

National Tube Co.

Nominal diam.	Actual diameters.		Thickness of wall.	Transverse area.		Threads per inch.	Length of perfect thread.	Weight per foot.
	External.	Internal.		External.	Internal.			
Ins.	Ins.	Ins.	Ins.	Sq. ins.	Sq. ins.		Ins.	Lbs.
⅛	0.405	0.269	0.068	0.129	0.057	27	0.19	0.24
¼	0.540	0.364	0.088	0.229	0.104	18	0.29	0.42
⅜	0.675	0.493	0.091	0.358	0.191	18	0.30	0.57
½	0.840	0.622	0.109	0.554	0.304	14	0.39	0.85
¾	1.050	0.824	0.113	0.866	0.533	14	0.40	1.13
1	1.315	1.049	0.133	1.358	0.861	11½	0.51	1.68
1¼	1.660	1.380	0.140	2.164	1.496	11½	0.54	2.27
1½	1.900	1.610	0.145	2.835	2.036	11½	0.55	2.72
2	2.375	2.067	0.154	4.430	3.356	11½	0.58	3.65
2½	2.875	2.469	0.203	6.492	4.780	8	0.89	5.79
3	3.500	3.068	0.216	9.621	7.383	8	0.95	7.58
3½	4.000	3.548	0.226	12.566	9.887	8	1.00	9.11
4	4.500	4.026	0.237	15.904	12.730	8	1.05	10.79
4½	5.000	4.506	0.247	19.635	15.961	8	1.10	12.54
5	5.563	5.047	0.258	24.301	19.986	8	1.16	14.62
6	6.625	6.065	0.280	34.472	28.890	8	1.26	18.97
7	7.625	7.023	0.301	45.664	38.738	8	1.36	23.54
8	8.625	7.981	0.322	58.426	50.027	8	1.46	28.55

Supports. — The method of support varies with the type of steam generator used. For horizontal boilers a series of cantilever brackets securely riveted to the shell forms a satisfactory solution. Fig. 174 gives the proportions of the ordinary pressed steel boiler bracket and its general location with regard to the center of the shell. To provide for expansion the rear brackets are supported on rollers. In order to minimize the overstraining of the shell in case the foundation settles, boiler brackets are arranged in groups of two. The location with reference to the length of the boiler is given in the lower figure. Whenever proximity to ring seams makes it impossible to locate the brackets as shown they should be placed symmetrically with respect to the tube sheets. Since the bulk of the boiler weight is suspended at the latter points it is well to place the brackets near the ends of the boiler.

Instead of supporting the boiler by brackets resting upon the foundation special hangers may be used to sling the boiler from structural steel beams above. Figs. 175 and 176 show two forms used for this purpose. The one at the left is made of boiler plate folded to shape and riveted to the shell. At the right is shown a cast steel hanger which is fastened in the same way. Heavy eye bolts are attached to the beams above by nuts and washers so that the level of the boiler may be adjusted as often as necessary. A total slope of one inch toward the blow-off pipe is generally given horizontal boilers.



DIMENSIONS AND LOCATION OF STANDARD STEEL BOILER BRACKETS

Fig. 174.

The support for vertical boilers consists of a circular ash-pit casting resting upon a concrete foundation. The water legs rest directly upon the ash-pit frame, its upper surface being finished for the purpose. Fig. 180, page 300, shows the details of this feature.

Scotch boilers are usually supported by two cradles of boiler plate or structural steel. The upper surface of the cradle corresponds to the curvature of the shell and the lower is bolted to a flat concrete foundation.

Locomotive type boilers are arranged to be transported on skids. Wooden or steel beams running along the sides of the boiler receive its weight at the rear end by means of steel brackets riveted to the outside furnace sheets. At the front end a Y-shaped support attached to the cylindrical shell or smoke box performs the same office. If the locomotive boiler is of considerable length oblong holes should be provided where the furnace brackets are bolted to the skids so that the increment of expansion will not strain the boiler shell. The same object may be accomplished by letting the cylindrical shell merely rest in the front support instead of bolting them together as described above.

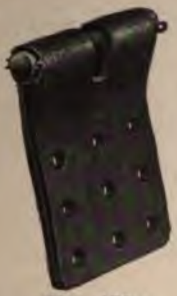


Fig. 175.

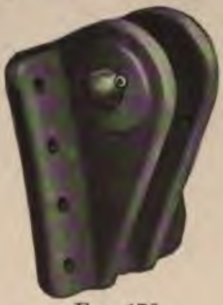


Fig. 176.

Grates and Ash-Pits. — While the design of the grate bars and setting should not be considered as part of the boiler design some data is necessary in this regard to determine various questions relating to combustion. It is very difficult to fire properly a stationary grate more than six and one-half feet long. The latter figure is generally considered the limit of length. Stationary grates for horizontal return tubular boilers generally vary in length by multiples of half feet. The proper width for a grate is the boiler diameter but when there is difficulty in obtaining sufficient grate area, a width 4 or 6 ins. greater than the diameter of the boiler is permissible. Grate bars are generally cast in multiple units measuring 3, 4 or 6 ins. in width, Fig. 177.*

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* The International Engineering Works.

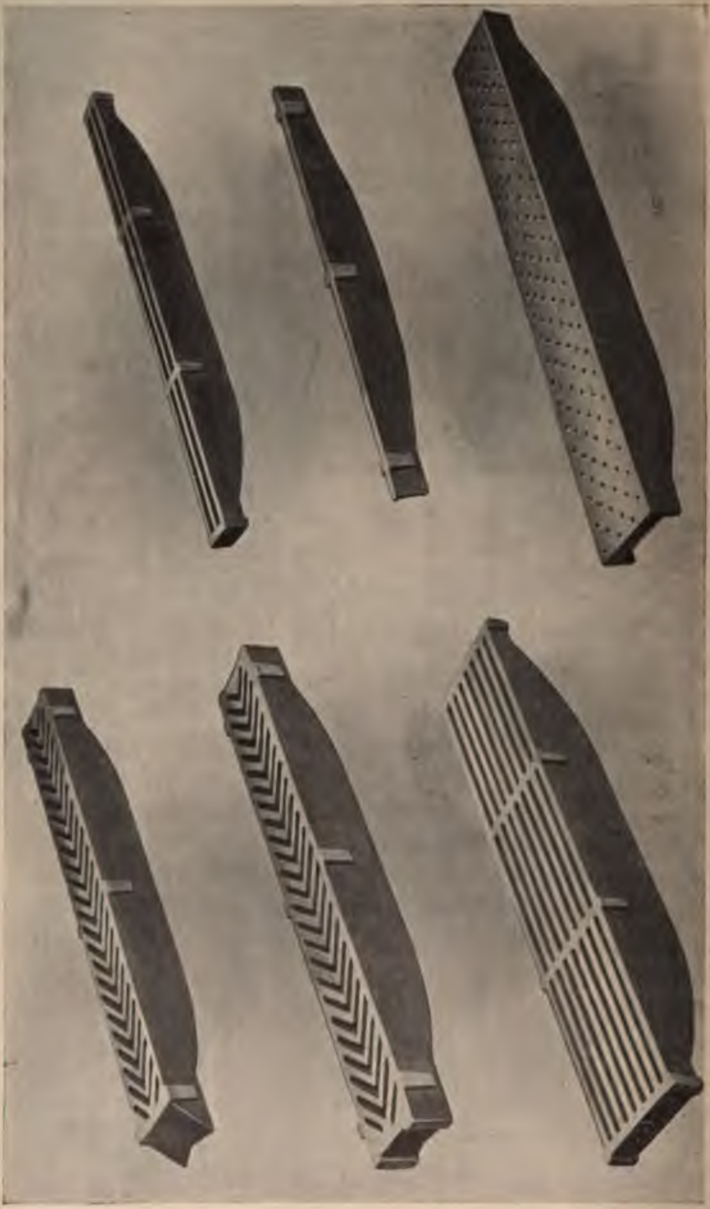

Fig. 177.

The character of the fuel has much to do with the form of the grate. The spaces between the grate bars ordinarily range from $\frac{1}{4}$ to $\frac{1}{2}$ in. in width, the latter being used for the coarser grades of coal. For burning bituminous slack mixed with the finer grades of anthracite, the air space is made even narrower than that given above. With forced draft the distance between grate bars may be but $\frac{1}{8}$ in. The air spaces are generally designed to aggregate from 30 to 45 per cent of the total grate area. Grates for burning chips and refuse coals are generally perforated slabs of cast iron, the diameter of the holes varying from $\frac{1}{4}$ to $\frac{3}{8}$ in. A raised boss of metal is sometimes left around the holes so that a thin layer of ashes will lodge on the surface of the grate. This arrangement prevents clinkers from adhering and clogging the draft holes.

In order to provide free discharge of ashes through the grate and prevent the sticking of clinkers between the bars all air spaces are made widely flaring in the direction of the ash-pit. Ledges of brick work must never be allowed to imprison ashes in the grate. When more than four feet in length grates are generally divided at the center, thus permitting freedom for expansion and avoiding the warping of the grate bars. Straight grate bars are sometimes ruptured by expansion. To avoid this difficulty the herring-bone grate shown in Fig. 177 has been devised. In this form the transverse ribs have opportunity for expansion without danger of rupture.

When the grate is made in two lengths the joint at the center is supported by a grate bearer resting upon the brick work at the sides of the ash-pit. This transverse beam is made up of two bars held together by thimbled bolts. The ends of the grate bars hook loosely over the bearer. The front end of the grate rests upon the dead plate which in turn is bolted securely to the boiler front. To permit expansion an inclined surface upon the dead plate is arranged to support the grate bars. At the bridge wall clearance room must be provided for the same purpose. In order that imprisoned ashes shall not heave over the bridge wall when the grate expands the bars are sometimes given an inclined end which will plough the ashes out of the way. Fig. 177 shows a herring-bone grate with its left end inclined for the purpose described above. The remaining figures show various blocks of straight grate bars cast together. With fine fuels the perforated form shown at the right in Fig. 177 is used.

Grates for Vertical Boilers. — The grates of vertical boilers are carried at the periphery of the furnace by a circular bearer bar either bolted to the mud-ring, as shown in Fig. 178, or supported by brackets projecting from the ash-pit frame. Such grates are designated as segmental or sector grates according to their form. Fig. 179 * shows typical grates for this purpose. A free expansion space at least ½ in. in width must be provided between the grate and furnace sheet. The sector form is illustrated in the lower part of the figure. Truncated sectors supported at their outer ends by the mud-ring and at the inner by a small circular center piece are arranged to fit the furnace loosely enough to provide the necessary room for expansion. The circular center is supported by a post and spider from the floor of the ash-pit.

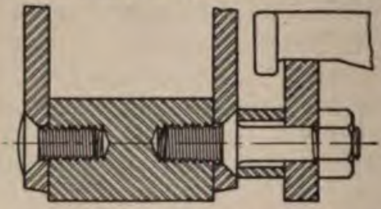
GRATE ATTACHMENT
TO
MUD RING
FIG. 178.

Fig. 180 illustrates the modern form of ash-pit casting for use with vertical boilers. It is frequently necessary to calk the inner and outer sheets where they join the mud-ring. Consequently the width of the ash-pit casting must be slightly narrower than the mud-ring. Projecting from the ash-pit casting are cantilevers upon which the grate bearer may rest.

The location of the ring seams in externally fired boilers is often determined by the length of grate and position of the bridge wall. Fig. 181 shows an assembling sketch which should be made in order to determine the relative positions of the grate, bridge wall and ring seams. Horizontal boilers are generally built with projecting uptake sheets since this effectually removes the dry plate from the region of the fire. The boiler front containing the fire doors is approximately one inch in thickness and comes just behind the front ring seam. To protect the boiler front from the fire a lining

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* The International Engineering Works.

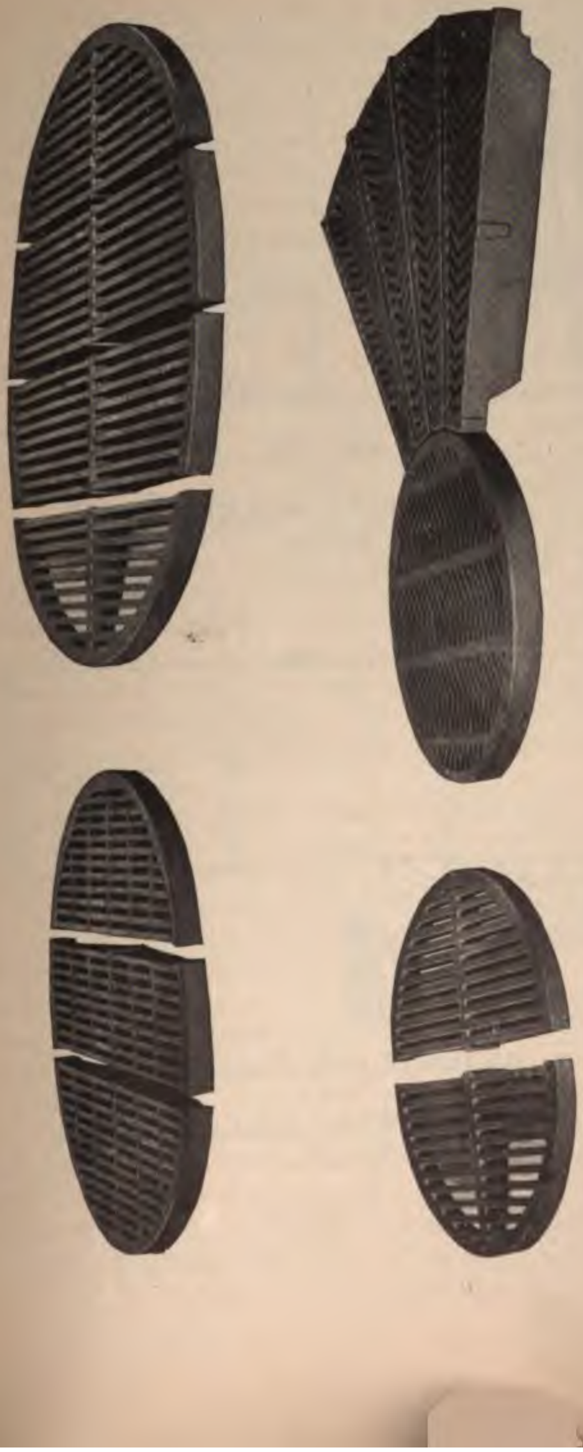
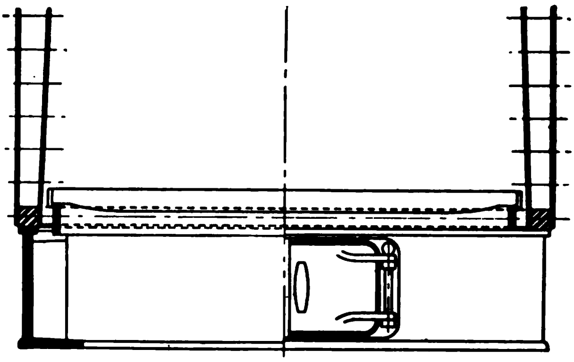
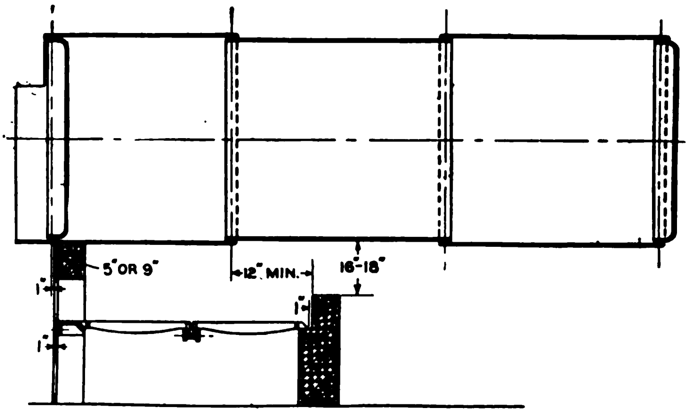


Fig. 179.



ASH-PIT
FOR
VERTICAL BOILERS
Fig. 180.



ASSEMBLING SKETCH
FOR
HORIZONTAL BOILERS
Fig. 181.

consisting of one or two rows of fire brick is interposed. For small boilers one row is sufficient but frequently two are employed. Beyond this the length of grate and expansion space should be laid off in determining the position of the bridge wall. To avoid the destructive action of the flames when impinging against a ring seam it is well to preserve a minimum distance of 12 ins. between the bridge wall and the girth seam. This stipulation frequently determines the length of the individual courses.

Fig. 182 * shows the usual method by which grates are installed in furnace tubes. In order to preserve the strength of the latter its walls must not be punctured by bolt holes except in close proximity to a ring seam. The dead plate which is part of the external furnace mouth is inserted from without and bolted securely to place by cast iron angles upon either side. These bolt holes through the walls of the furnace tube comprise the only perforations allowed. Since they are so close to the flange seam, where the front tube sheet and furnace join, they do not seriously weaken the corrugations. The dead plate consists of a slab of cast iron with a sloping edge upon which the grate bars rest. Cramping due to expansion is thus avoided by allowing the grate bars to slide up the incline. In order to prevent ashes and coal from wasting at the sides of the grate in the corrugation spaces, special grate bars are designed for these localities. The contour of the corrugations is roughly approximated by the grate bar, sufficient room being allowed for expansion.

At the center of the grate is a bearer bar made up by bolting two beams of rectangular section together with thimbled bolts. At either end of the bearer is a U-shaped clip in which the ends rest. This clip is bolted to a semi-circular ring of cast iron which rests in the adjacent corrugation. Thus the grate is supported at its center without puncturing the furnace tube wall.

The bridge wall consists of a hollow semi-circular casting. In its front side is a sliding door to admit air. Its rear side is perforated with many small holes to spread the air currents as they pass into the combustion chamber. The top of the bridge wall is provided with a sloping surface similar to that employed upon the dead plate. Two or three courses of fire brick are laid over the bridge wall casting to protect it from the flames. The fire brick are held in place by a slab bolted to the rear of the bridge wall. To prevent the bridge wall from overturning, bolted clips are

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* The Continental Iron Works.

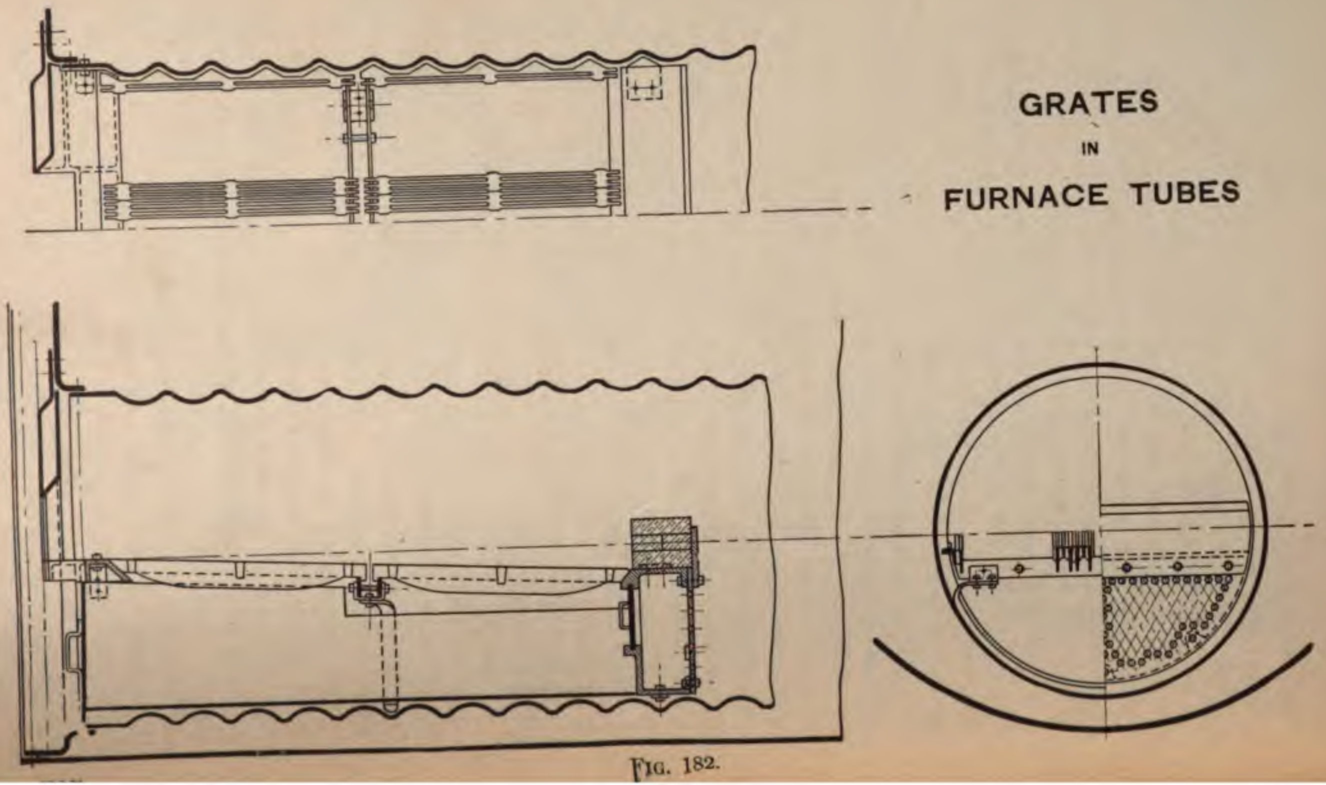

FIG. 182.

COMBUSTION CHAMBER
IN
DRY BACK SCOTCH BOILERS

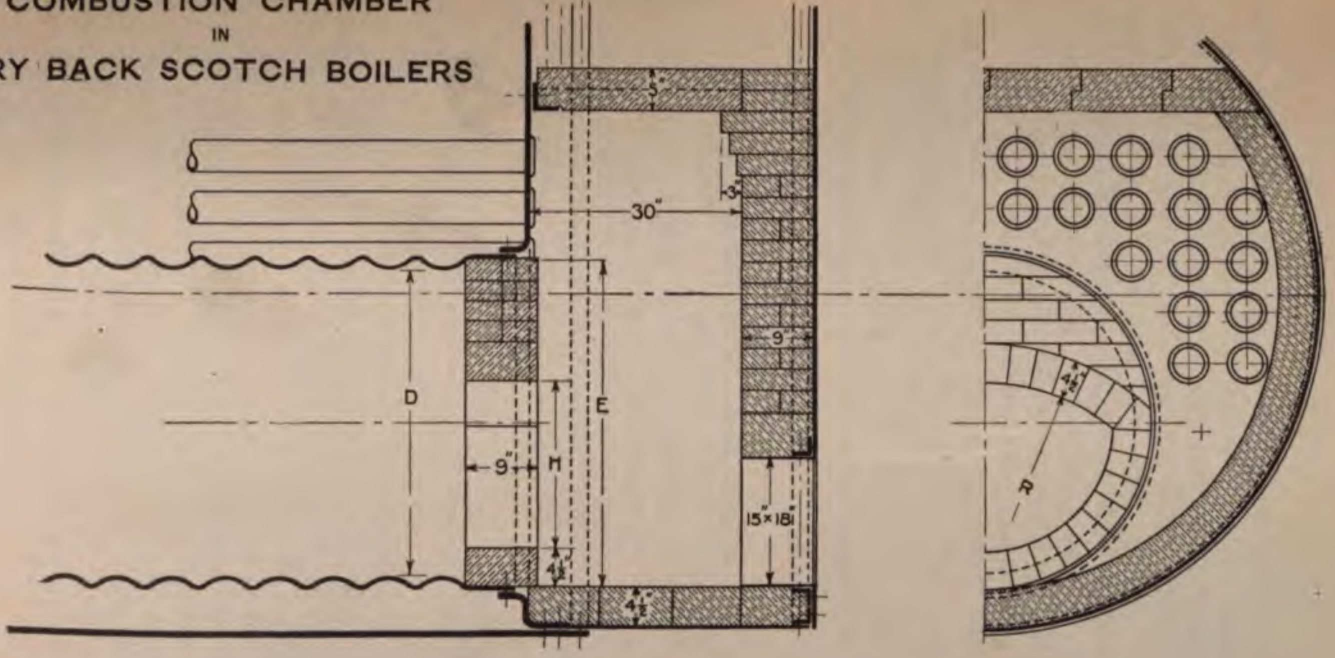

FIG. 183.

provided at either side and the bottom, the latter fitting into the corrugations with fair snugness. Clearance is provided for expansion in all the bolt holes and fastenings throughout this arrangement. Grates of this type slope from 4 to 6 ins. toward the rear when of ordinary length. To facilitate the removal of ashes a semi-circular ash pan sheet is fitted underneath the grate.

In dry back Scotch boilers the flames are returned to the tubes by means of a combustion chamber, Fig. 183,* built of fire brick and located at the rear in an extension of the shell.

The shell extension is usually made of $\frac{1}{4}$ in. steel plate riveted to the main shell of the boiler. A circular ring of bent angle bar is riveted to the end of the combustion chamber shell. The flat circular end of the combustion chamber is made of $\frac{3}{16}$ in. steel plate bolted in place in the form of segmental panels. To stiffen this end plate against the effects of heat, transverse angles are bolted at frequent intervals. The panels are easily removable to permit repair of the brick work within. An inverted arch of fire brick about $4\frac{1}{2}$ ins. thick is first sprung around the shell extension as high as the tops of the tubes. At the rear a fire brick wall 9 ins. thick is securely laid in place with a corbelled ledge near its top upon which the roof tiles rest. The latter consist of special fire brick slabs 5 ins. thick, 15 ins. wide and 30 ins. long. Their front ends are supported upon an angle riveted to the rear boiler head just above the top row of tubes. To facilitate cleaning, a door is

TABLE LIII.

PROPORTIONS OF BAFFLE ARCHES

for use in

MORISON FURNACE TUBES.

For significance of letters see Fig. 183.

Inside furnace diam, *D*.	Inside diam. of furnace end, *E*.	Radius of arch, *R*.	Height of opening, *H*.
Ins.	Ins.	Ins.	Ins.
36	39	$24\frac{1}{2}$	20
38	41	26	$21\frac{1}{2}$
41	44	28	$23\frac{1}{2}$
45	48	31	$26\frac{1}{2}$
50	53	$34\frac{1}{2}$	30

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* The Continental Iron Works.

oted

10 pp missing

and steam nozzle. Thus it is found that the center line of the joint should be $8\frac{1}{2}$ ring seam pitches from the top of the boiler. The middle course joint must be placed to clear the manhole ring. It is found necessary to place it $11\frac{1}{2}$ pitches from the top of the boiler.

Uptake. — Taking the cross-sectional area of the uptake one-eighth the grate area,

$$\text{Area of uptake} = \frac{30.25}{8}$$
$$= 3.78 \text{ sq. ft. or } 544 \text{ sq. ins.}$$

Using as a mean value for the width 0.69 of the diameter, it will be

$$0.69 \times 66 = 45.5 \text{ ins.}$$

The length corresponding is

$$\frac{544}{45.5} = 11.96 \text{ ins.}$$

Make uptake 46 ins. long by 12 ins. wide.

This width is found to be satisfactory with respect to the placing of the longitudinal joint.

Manhole and Steam Nozzle Riveting. — The arrangement of rivets in the manhole seam is that shown in Fig. 156, p. 276. Steam nozzles of specified size are located near the center of the front and rear courses. Rivets $\frac{7}{8}$ in. in diameter are used in both cases.

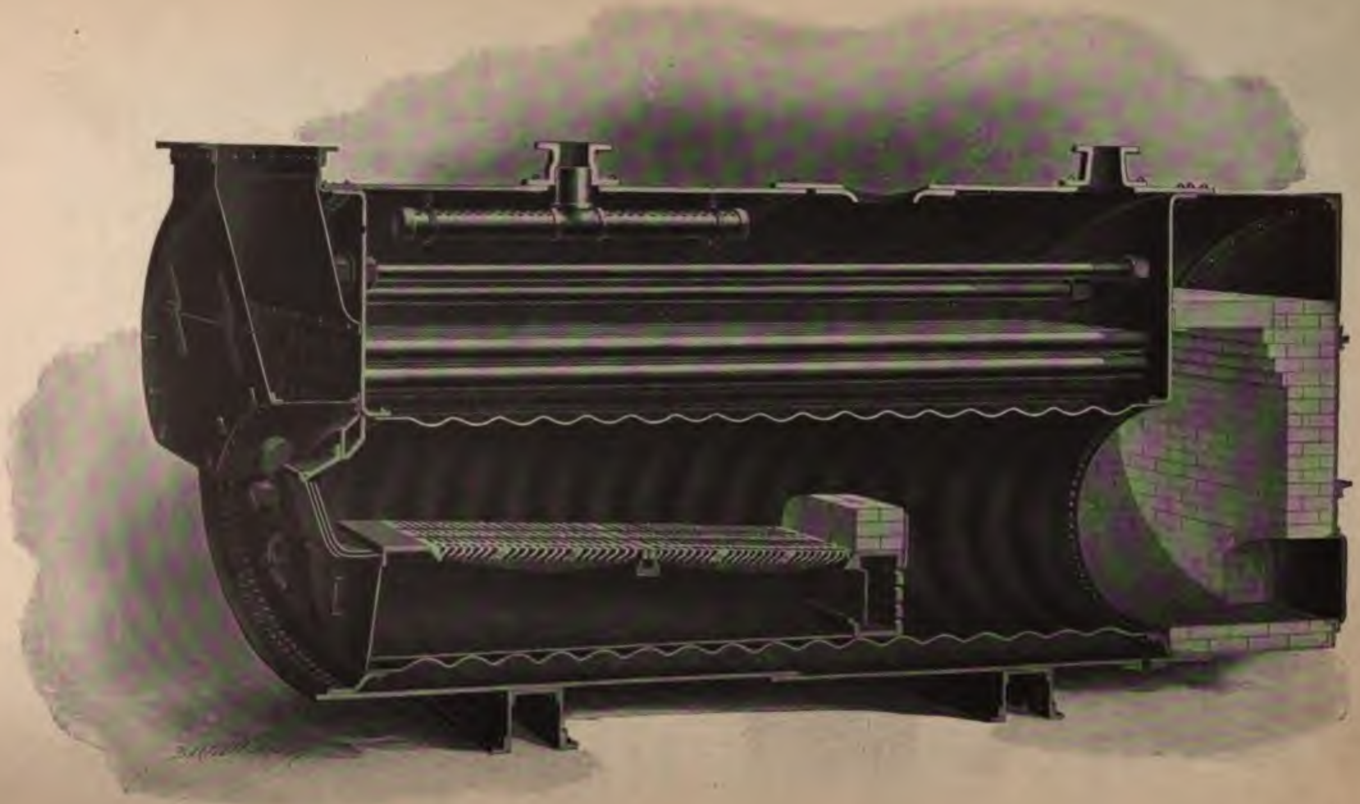
Fusible Plug. — In order that the fusible plug, Fig. 169, p. 286, may be located 2 ins. above the tubes it is necessary to bend up slightly the lower flange of the channel.

Feed Pipe Bushing. — The feed pipe bushing, Fig. 170, p. 288, is located at the center of the lower channel near the left end.

Blow-off Connection. — A forged flange, Fig. 165, p. 282, is attached at the bottom of the boiler 6 ins. from the rear ring seam.

Brackets. — Eight brackets of the type shown in Fig. 174, p. 294, are located in accordance with the discussion of Art. 69, p. 293.

The following drawings, Figs. 185 and 186, have been prepared to accompany the foregoing calculations.



DRY BACK SCOTCH BOILER.
The Bigelow Co.
FIG. 187.

CHAPTER VI.

DESIGN OF A DRY BACK SCOTCH BOILER WITH COMBUSTION CHAMBER IN SHELL EXTENSION.

In this chapter the complete design of a dry back Scotch boiler will be worked out as an application of the principles laid down in the foregoing discussion.

The illustration, Fig. 187, shows the usual form in which this boiler is manufactured. Partaking as it does of many of the characteristics of the marine Scotch boiler, it serves as a compact and efficient steam generator. It is easily transportable, requiring no other setting than a pair of structural steel cradles and the usual covering of insulating material. The outer shell, consisting generally of two courses, can be safely made of sufficient diameter and thickness to comprise the necessary internal heating surface while confining boiler pressures of considerable intensity. To give transverse strength to the shell the ring seams are usually double-riveted lap joints. The longitudinal seams permit as much complication as is necessary to gain the requisite efficiency of joint. A corrugated furnace tube securely riveted to flanges in both of the tube sheets forms a receptacle for a long and rather narrow grate and provides at the same time heating surface of great value. The furnace seams must be of the single-riveted lap type in order to withstand the intense heat. Even when so designed, however, their durability is not great and a baffle arch is frequently located, where the furnace joins the rear tube sheet. This expedient assists in mingling the products of combustion and protects the ring seam from overheating. The grate and bridge wall are fastened in the furnace tube without puncturing the walls of the latter except in the case of two bolts close to the fire door. Since the ash-pan has but little value as heating surface the grates are given a decided slope downward toward the bridge wall.

In an extension of the shell at the rear is placed an external combustion chamber built of fire brick with special slabs forming the roof. This brick work, heated to incandescence, increases the vigor of the combustion and returns the hot gases to the mouths of the boiler tubes. In the larger sizes of boilers two furnace tubes may be used communicating with a common combustion chamber. The furnaces fired in alternation thus assist one another in the consumption of smoke.

The remaining space in the tube sheets below the limit of the water level is utilized for the insertion of boiler tubes. A large circulation space is provided over the furnace tubes and the individual spaces between tubes are kept as large as possible consistent with obtaining the requisite heating surface.

The upper portions of the tube sheets are not exposed to the products of combustion, hence comparatively thick plate may be used for these members and correspondingly heavy construction may be employed in the staying system. The latter consists of through rods carrying heavy check-nutted washers at their ends. The washers distribute the holding power of the rods over a large area of plate so that diagonal stays are not necessary. Light angles are riveted externally to both the front and rear tube sheets in order to provide for the respective attachment of the uptake and combustion chamber roof. To give access to the exterior of the furnace tubes commodious handholes are provided at each side. In the case of double furnace tubes additional manholes are generally installed in the triangular spaces both above and below. A manhole in the top of the shell near the rear provides for the inspection and repair of the stay rods. Since a fairly high water level is carried in boilers of this type the withdrawal of steam is generally accomplished by use of a perforated dry pipe. A second steam nozzle is riveted to the shell to provide for the attachment of the safety valve. The end sheets in the combustion chamber extension are bolted to place so that the brick work is entirely accessible for repair.

73. Specifications. — The following general specifications are intended to apply to the design of any dry back Scotch boiler.

Shell Plates. — Plates shall be of best quality O. H. Firebox Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	52,000
Not more than	62,000
Yield point, min. lbs. per sq. in.	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T.S.}}$

Heads. — Heads shall be of best O. H. Flange Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	55,000
Not more than	65,000
Yield point, min. lbs. per sq. in.	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T.S.}}$

All heads shall be machine-flanged by the spinning process to an inner radius not less than two times the thickness, and thoroughly annealed.

Rivets. — Rivets shall be of the best quality of soft steel according to the following specifications:

Tensile strength, lbs. per sq. in.:	
Not less than	45,000
Not more than	55,000
Elongation in 8 ins., min. per cent, but need not exceed 30 per cent	$\frac{1,500,000}{\text{T.S.}}$
Shearing strength, lbs. per sq. in	45,000
Crushing strength (bearing pressure), lbs. per sq. in. of projected area	96,000

Rolling. — The plates shall be rolled cold by gradual and regular increments to the exact radius required and the whole circumference rolled to a true circle. Butt-straps shall be rolled to the same radius as the shell in special forms made for that purpose.

Planing. — The edges of the plates shall be beveled to an angle of about 75 degrees on a plate planing machine. After the heads have been flanged and annealed, the calking edge shall be turned off on a milling machine to the same bevel as the shell plates.

Seams and Riveting. — The longitudinal seams shall be of the butt-joint type with inside and outside cover plates. The circular or girth seams in the shell shall be of the double-riveted lap joint type; those at the ends of the furnace of the single-riveted lap joint type. Longitudinal seams shall come well up in the steam space of the boiler and shall break joints in the several courses.

Rivets shall be $\frac{1}{16}$ in. smaller in diameter than the holes they are intended to fill. All holes shall be drilled to size with all plates and cover plates in place, after the plates have been rolled. After drilling the holes the plates shall be taken apart and the burrs removed from the edges of the holes. Previous to the drilling the holes may be punched cold to a diameter $\frac{1}{4}$ in. less than that to which they are to be finished. All riveting shall be done with a hydraulic riveting machine wherever practicable, pressure being kept on each rivet until it has taken its shrinkage to insure tight joints. Rivet heads on all machine-driven rivets shall be of the type ordinarily used in pressure work. Where necessary, the rivets may be pneumatically driven, the formed head taking the above shape, the other retaining its original shape.

Calking. — All seams shall be carefully calked with a pneumatic hammer, using a round-nosed calking tool.

Tubes. — Tubes shall be made of the best American lap-welded or seamless steel, arranged in vertical and horizontal rows with a wide central space for circulation over the furnace tube.

Furnaces. — The boiler shall be provided with one corrugated furnace tube of the Morison suspended type securely riveted to flanges in the front and rear tube sheets. The inside diameter shall be sufficient to secure the requisite grate area consistent with a reasonable length. The thickness of the furnace tube shall be such as to make it conform to the rules of the U. S. Board of Supervising Inspectors.

Combustion Chamber. — The boiler shall be provided at its rear end with a combustion chamber 30 ins. deep in the clear. The rear wall of the combustion chamber shall be 9 ins. thick, of fire brick, the sides and bottom formed by laying an inverted fire brick arch $4\frac{1}{2}$ ins. thick inside the dry back extension. The top of the combustion chamber shall be made of fire brick tile slabs 5 ins. thick, 30 ins. long and 15 ins. wide, with projections at the sides so as to interlock with one another. These slabs shall rest at their rear ends upon a corbeled-out portion of the combustion chamber wall and at their front ends upon a 3 ins. × 3 ins. × $\frac{5}{16}$ in. angle riveted to the rear boiler head just above the top row of tubes. The fire brick lining may be chipped locally to accommodate stay rod washers and an occasional tube. The shell of the combustion chamber shall be made of $\frac{1}{4}$ in. boiler plate fastened by a single-riveted lap joint to an extended portion of the main shell. The frame for this chamber shall be made of $2\frac{1}{2}$ ins. × $2\frac{1}{2}$ ins. ×

$\frac{5}{16}$ in. angles, with the lower plates riveted and the upper ones bolted to place, the latter to give access to the cavity above the tile slabs. A door 18 ins. wide by 15 ins. high shall be located at the bottom of the combustion box and a liner $\frac{1}{4}$ in. thick shall be riveted to the angles forming the door frame.

Staying above Tubes. — Each head above the tubes shall be stayed and the arrangement shall be such as to secure an even spacing of the stayed points and a good support for the entire surface. The reaches of plate between stays in the tube sheet shall not exceed in length those allowed for stay bolts on flat plates of similar thickness. The stay rods shall be so spaced as to secure the most uniform loading possible. Through stay rods shall be of weldless steel type with upset threaded ends; they shall conform to the dimensions and specifications usually given for such members and shall pass through and be secured to the heads with standard nuts. Heavy steel boiler plate washers with beveled edges shall be used on the outside to distribute the stress and check-nuts of half standard thickness shall be used on the inside of the boiler.

All rods above tubes shall be located far enough apart horizontally so that a man may readily enter the boiler through the manhole.

Staying below Tubes. Whenever necessary, each head below the tubes shall be braced at the sides by one or more through rods of a type similar to that specified above. The front end of the stay rods shall be upset and threaded. The rear end shall be upset and forged to the form of an eye and fastened by means of a pin to two steel angles securely riveted to the rear head. The angles shall be set off from the head at least 3 ins. by pipe thimbles on the rivets so that the metal in the angles shall not be subjected to the heat of the fire. The front end of the rods shall pass through and be secured to the front head by inside and outside nuts, and heavy external beveled washers. When a handhole is in close proximity to a stay rod nut in the front head, the washer may be incorporated with the reinforcing pad around the handhole and placed inside. In such a case the pad shall be riveted securely to the tube sheet by rivets having heads countersunk on the inside.

Manhole in Shell.— One manhole 11 ins. × 15 ins. shall be located in the shell on top of the boiler, in the center of the rear course.

It shall have a pressed steel frame or ring securely riveted to the shell. It shall be fitted with a pressed steel cover, bolt and yoke, and a copper or rubber gasket. . The frame shall have a net cross-sectional area on a line through its center parallel to the axis of the shell, the tensile strength of which is at least equivalent to that of the shell plate removed on the same line. The frame shall be riveted to the shell with two rows of rivets. The resistance of the rivets in shear or in crushing on one side of the longitudinal section through the center of the frame shall not be less than the tensile resistance of the plate removed on the same section.

Handholes. — Two 5½ ins. × 8 ins. handholes shall be located in the lower part of the front head, one each side of the furnace. There shall be at least 2 ins. of clear plate all around each handhole, the same to be reinforced by a pad or ring of the same thickness as the head and at least 2 ins. in width. The ring shall be securely riveted to the tube sheet on the inside with the rivet heads countersunk and left smooth to form a seat for the handhole gasket. Provision shall be made for the use of a standard pressed steel handhole cover and yoke.

Nozzles. — Two cast or forged steel nozzles shall be machine-riveted to the shell for main steam and safety valve connections. The flanges which join the shell shall be beveled and calked on the outside.

Water Column Connections. — Two holes of diameter specified shall be tapped for water column connections. The water connection shall be taken from a point in the right side of the shell near the horizontal diameter at least 8 ins. back from the front tube sheet and the steam connection from a similar point at the top of the shell.

Feed Pipe Connections. — Upon the left side of the boiler at a level a little below that of the top row of tubes and about one foot from the front tube sheet, a forged steel pipe flange shall be riveted to the shell. The size of the flange shall be sufficient to accommodate a suitable feed pipe bushing. The feed pipe shall be screwed into this bushing from without and a male and female elbow with distributing pipe from within. The latter shall extend about 4 ft. along the side of the boiler toward the rear and shall be perforated on its lower side to throw the feed water into the circulation.

Blow-off Connections. — A forged steel pipe flange shall be riveted to the bottom of the boiler near the rear end and tapped to receive the size of blow-off pipe specified.

Smoke Bonnet. — Provision shall be made by riveting $2\frac{1}{2}$ ins. × $2\frac{1}{2}$ ins. × $\frac{5}{16}$ in. angles across the front head just above the tubes and around the furnace opening for the attachment of a sheet-iron smoke bonnet.

Fusible Plug. — A fusible plug of approved pattern, filled with pure tin, shall be located in the rear head of the boiler not less than 2 ins. above the top of the upper row of tubes.

Test. — A hydrostatic pressure test (cold water) of one and one-half times the working pressure for which the boiler is intended, shall be applied, and the boiler shall be suitably designed for tight joints under that pressure.

74. Statement of Problem. — The solution of the following problem will be given in full.

Design a dry back Scotch boiler with combustion chamber in shell extension, to conform to the preceding specifications and to embody the following general dimensions:

General Dimensions:

Rated Horse-Power (A.S.M.E. standard)	100
Working Pressure, lbs. per sq. in	150
Length of Tubes, ft	$12\frac{1}{2}$
Number of Courses	2
Number of Furnaces	1
Diameter of Steam Nozzles (2), ins	5
Diameter of Feed Pipe, ins	$1\frac{1}{4}$
Diameter of Blow-off Pipe, ins	2
Diameter of Water Column Connections, ins	$1\frac{1}{4}$
Kind of Coal	Bituminous
Types of Joint to be used:	
Ring Seam Joints	Double-Riveted Lap.
Furnace Joints	Single-Riveted Lap.
Longitudinal Joints	Butt-joint with inside and outside cover plates, having an efficiency of at least 94 per cent.

Design to include:

(*a*) Complete calculations.

(*b*) Working drawings fully dimensioned, of

(1) Tube sheet, with staying. Scale, 3 ins. = 1 ft.

(2) Boiler; end view, longitudinal section, development of joints. Scale, $1\frac{1}{2}$ ins. = 1 ft.

The following specific values of constants, for which a general discussion is given in Chap. IV, will be assumed.

Rate of Evaporation, lbs. of water per lb. of coal, about....	9.5
Rate of Combustion (bituminous coal), lbs. per sq. ft. of Grate Surface per hr., about..........................	18
Ratio: Length to Outside diam. of Tubes, about...........	45 to 1
Ratio: Internal Transverse Tube Area to Grate Area, about	1 to 6
Steam Space per Boiler H.P., cu. ft., at least..............	0.70
Ratio: Steam Space to Water Space, about...............	1 to 3
Heating Surface per Boiler H.P., sq. ft., at least...........	6
Ratio: Heating Surface to Grate Surface, about...........	34 to 1
Boiler Diameters to vary by increments of, ins...........	6
Reach of Riveting Machine, ft..........................	7
Grate: Maximum Length, ft............................	6½
Width equals internal diameter of furnace tube.	
Length may vary in increments of, ins............	3 or 4
Least Factor of Safety..................................	5

75. Calculations.

Grate. — The number of pounds of water to be evaporated per hour may be obtained from the rated horse-power (A.S.M.E. standard, p. 5).

$$100 \times 34.5 = 3450 \text{ lbs. of water per hour.}$$

The coal to be burned per hour is found from the above and the rate of evaporation given.

$$\frac{3450}{9.5} = 363 \text{ lbs. of coal per hour.}$$

From the rate of combustion the theoretical grate area is found to be

$$\frac{363}{18} = 20.18 \text{ sq. feet.}$$

Tubes. — The tube diameter for the given ratio of length to outside diameter is

$$\frac{12.5 \times 12}{45} = 3.33 \text{ ins.}$$

Tubes 3.5 ins. in diameter will be used.

From Table XXIX, p. 190,

Internal circumference = 10.24 ins.,
External transverse area = 9.62 sq. ins.,
Internal transverse area = 8.35 sq. ins.

The total internal transverse tube area is computed from the theoretical grate area and the ratio given.

$$\frac{20.18 \times 144}{6} = 484.3 \text{ sq. ins.}$$

The number of tubes to give this area is

$$\frac{484.3}{8.35} = 58.0.$$

The best possible arrangement of this number of tubes will therefore be laid out upon the drawing.

Furnace Size. — Try a grate 6 ft. in length. Corresponding width of grate will be

$$\frac{20.18 \times 12}{6} = 40.36 \text{ ins.}$$

Use grate 40 ins. wide. Assumed mean diameter of furnace tube may then be taken as

$$40 + 2 = 42 \text{ ins.}$$

Furnace Tube Thickness. — Using Supervising Inspectors' Rule for Morison tubes, p. 207,

$$\begin{aligned} t &= \frac{42 \times 150}{15{,}600} \\ &= 0.399 \text{ in.} \end{aligned}$$

Use furnace tube thickness $\frac{7}{16}$ in. From the standard proportions of Morison tubes, Fig. 117, p. 205, the actual outside diameter of the corrugations will be

$$40 + \tfrac{7}{8} + 3 = 43\tfrac{7}{8} \text{ ins.}$$

and the actual mean diameter will be

$$\frac{40 + 43\frac{7}{8}}{2} = 41\tfrac{15}{16} \text{ ins.}$$

The actual outside diameter of the plain portion at the ends will be

$$43\tfrac{7}{8} + \tfrac{1}{4} = 44\tfrac{1}{8} \text{ ins.}$$

Boiler Diameter. — The total volume of the boiler will be, from the original ratios,

Steam space, $100 \times 0.7 = 70.0$ cu. ft.
Water space, $70.0 \times 3 = 210.0$ cu. ft.

Tube space, $\dfrac{58 \times 9.62 \times 150}{1728} = 48.44$ cu. ft.

Furnace volume, $\dfrac{\pi\,(41.94)^2 \times 150}{4 \times 1728} = 119.9$ cu. ft.

Total 448.34 cu. ft.

For a length of 12.5 ft. the corresponding area of the head will be

$$\frac{448.34 \times (12)^2}{12.5} = 5165 \text{ sq. ins.}$$

The latter figure corresponds to a diameter of 81.10 ins. To bring the diameter to a practical figure and make some allowance in the capacity of the boiler the diameter will be made 84 ins.

Water Levels. — The normal mean water level is calculated on the basis of transverse areas.

The area of an 84 in. circle is 5541.8 sq. ins.

The external area of 58 tubes, 3.5 ins. diameter, is

$$58 \times 9.62 = 558.0 \text{ sq. ins.}$$

The transverse area of the furnace tube at its average external diameter is, Fig. 117,

$$\frac{\pi}{4}(43.88 - 1.5)^2 = 1410.3 \text{ sq. ins.}$$

The total area to be subtracted from the transverse area of the shell will be

$$558.0 + 1410.3 = 1968.3 \text{ sq. ins.}$$

The net area for steam and water will then be

$$5541.8 - 1968.3 = 3573.5 \text{ sq. ins.}$$

Of this one-fourth is steam and three-fourths is water. Therefore the steam area equals

$$\frac{3573.5}{4} = 893.4 \text{ sq. ins.}$$

To determine the height of the segment

$$\frac{A}{D^2} = \frac{893.4}{(84)^2} = 0.1266.$$

The corresponding ratio from the plot, Fig. 118, p. 213, is

$$\frac{h}{D} = 0.219,$$

and

$$h = 18.40 \text{ ins.}$$

This is the normal mean water level and the distance to the tops of the tubes will be, according to Table XXXVII, on p. 214,

$$18\tfrac{3}{8} + 3 + 3\tfrac{1}{2} = 24\tfrac{7}{8} \text{ ins.}$$

or

$$42 - 24\tfrac{7}{8} = 17\tfrac{1}{8} \text{ ins.}$$

above the center of the boiler.

The maximum level of the tubes to provide for a net steam volume of 0.7 cu. ft. of steam per horse-power is next found as follows: The area of the steam segment necessary to accommodate 70 cu. ft. of steam in a boiler 12.5 ft. in length is

$$\frac{70 \times 144}{12.5} = 806.1 \text{ sq. ins.}$$

Then

$$\frac{A}{D^2} = \frac{806.1}{(84)^2}$$
$$= 0.1142.$$

The corresponding ratio from the plot, Fig. 118, p. 213, is

$$\frac{h}{D} = 0.203,$$

whence

$$h = 17.05 \text{ ins.},$$

giving a minimum distance to the tube tops of 23.55 ins.

Thickness of Shell and Tube Sheet. — The desired shell thickness is readily obtained with the given data from the plot, Fig. 11, p. 57, where the joint efficiency is taken as 94 per cent.

$$\text{Thickness} = 0.609 \text{ in.}$$

Use a thickness of shell equal to $\frac{5}{8}$ in. The thickness of tube sheet will be assumed $\frac{11}{16}$ in.

Tube Sheet Layout. — Before proceeding farther with the calculations, the arrangement of tubes must be determined from the tube sheet drawing.

Locating the furnace tube as shown in Fig. 188, the general arrangement in regard to circulation and spacing is obtained.

It is seen that there is sufficient room below the normal tube level for the placing of but 56 tubes with a circulation space 8 ins. wide above the furnace. This is probably near enough the theoretical number to give good results.

Final Ratios. — Having established the steam and water volumes of the boiler, the ratios called for upon the tube sheet drawing, Fig. 188, will next be determined.

The grate will be assumed 3 ft. 4 ins. wide by 6 ft. long, giving an area of 20 sq. ft.

The heating surface inside of the tubes is

$$\frac{56 \times 10.24 \times 150}{144} = 597.3 \text{ sq. ft.}$$

The ash-pan sheet in the furnace tube has no special value as heating surface and hence will be excluded in the calculation. Taking the upper semi-circumference of the furnace tube as far back as the rear of the bridge wall and the entire circumference thereafter as valuable heating surface, the area, based on the average diameter, is computed as follows:

Width of bridge wall equals 10 ins., width of dead plate equals 7.5 ins., Fig. 182, on p. 302.

Total distance to rear of bridge wall is, therefore,

$$72 + 10 + 7.5 = 89.5 \text{ ins.}$$

Length of furnace tube beyond bridge wall is

$$150 - 89.5 = 60.5 \text{ ins.}$$

The heating surface inside of the furnace tube computed from this data is therefore 96.42 sq. ft.

The rear tube sheet as far up as the tube tops is very valuable heating surface since it is in direct contact with the hot gases of the combustion chamber. Subtracting the total transverse area of the tubes and furnace from the segment thus exposed to the products of combustion, the area remaining is 15.23 sq. ft.

The total heating surface is, therefore,

Tubes	597.3 sq. ft.
Furnace	96.42 sq. ft.
Rear head	15.23 sq. ft.
Total	708.95 sq. ft.

This quantity divided by the horse-power of the boiler indicates 7.10 sq. ft. of heating surface per horse-power.

Taking the original grate surface as 20 sq. ft. the ratio

$$\frac{H}{G} = \frac{708.95}{20}$$
$$= 35.45,$$

which is close to the ratio specified in the problem.

The internal transverse area of the tubes is

$$\frac{56 \times 8.35}{144} = 3.25 \text{ sq. ft.}$$

and the ratio

$$\frac{C}{G} = \frac{3.25}{20}$$
$$= \frac{1}{6.15},$$

which closely approximates the data.

Since the tubes are arranged to correspond exactly to the normal mean water level, the ratio of steam space to water volume or

$$\frac{S}{W} = \frac{1}{3},$$

which was the original assumption.

The volume of steam per horse-power should be based upon the actual level at which the tubes are placed. The segmental steam area as used in the calculation of the normal water level upon p. 336 was 893.4 sq. ins. Then the volume of steam per horse-power is

$$\frac{S}{H.P.} = \frac{893.4 \times 150}{1728 \times 100}$$
$$= 0.776 \text{ cu. ft.},$$

a figure well in excess of the stipulated amount.

Disengaging Surface per Horse-Power. — With the tubes located in correct relation to the mean water level, the value of

$$\frac{h}{R} = \frac{18.38}{42}$$
$$= 0.438.$$

Then from the plot, Fig. 119, p. 216,

$$\frac{w}{R} = 1.653,$$

or $$w = 69.43 \text{ ins.}$$

The net disengaging area per horse-power is, then,

$$\frac{69.43 \times 12.5}{12 \times 100} = 0.723 \text{ sq. ft.}$$

This figure is evidently sufficient as compared with the data given in Table XXXVIII, p. 215.

Staying. — With an assumed thickness of tube sheet of $\frac{11}{16}$ in. the maximum allowable plate spacing from Table XXXIX, p. 233, is 8.83 ins. Drawing a series of mean radials as explained in Fig. 137, p. 245, the medial line of the unstayed plate is determined. Assuming the washers 8 ins. in diameter and of the same thickness as the tube sheet, the arrangement shown in Fig. 188 is found by experiment. The line *op* at the side indicates the limit to which the plate is inherently stiff. The washers are then spaced along the medial line so as to give approximately equal reaches of plate between their respective stiff circles and the rigid plate at the sides. The plate stayed by each washer is assigned by the method of loci explained in Fig. 139, p. 248. The larger areas held by the upper rods measure by planimeter 193.7 sq. ins. each of which corresponds to a gross load of 29,060 lbs. Referring to Table XLIV, p. 246, it is seen that a stay rod with a shank $2\frac{1}{4}$ ins. in diameter is required. This diameter will be used for all the stay rods in the upper portion of the boiler. This size of rod calls for an upset diameter of $2\frac{5}{8}$ ins. and carries a $2\frac{3}{4}$ in. standard hexagon nut measuring $4\frac{1}{4}$ ins. across the flats. The drawing indicates the appearance of the staying with the above proportions.

The diagonal of a square having the maximum plate spacing, 8.83 ins., for its side measures 12.49 ins. By trial upon the drawing it is found that no point in the stayed surface is more than half this amount distant from a point of support.

The staying in the lower portion of the tube sheet has to be worked out by trial. The stiff lines for tube sheet and furnace flanges are indicated in Fig. 188. Mean radials to these two circles will run approximately to the point *c* midway between the centers *a* and *b*. By trial a position *cg* is found where the intercept *fg* cut off by the circles is equal to 8.83 ins., the maximum allowable plate spacing. Below this line the tube sheet will take care of itself but staying must be employed above. The stiff line of the tubes drawn in the usual manner defines the upper limit of the unstayed plate.

Let it be assumed that four rivets $\frac{15}{16}$ in. in diameter are used, thimbles of 1 in. pipe being inserted upon their shanks. The width of the eye at the end of the stay rod may be assumed to be $1\frac{3}{8}$ ins. which will probably be sufficient for this case.

Assuming two angles, $3\frac{1}{2}$ ins. × 5 ins. × $\frac{1}{2}$ in., the necessary spacing between rivets would be about $4\frac{1}{2}$ ins. From the proportions of the area to be stayed the other dimension between rivets should be about 6 ins. Having drawn the arrangement as shown in Fig. 188, the plate to be allotted to each rivet may be defined by the method of loci explained in connection with Fig. 139, p. 248. The total area supported by the four rivets equals 78.84 sq. ins. made up of 22.64, 19.64, 19.28 and 17.28 sq. ins. upon the respective rivets. The area which a $\frac{15}{16}$ in. rivet will safely sustain from Table XL, p. 235, is 27.62 sq. ins., hence all the above rivet loads are safe ones. The total rod load corresponding to the above area is 11,830 lbs. By reference to Table XLIV, p. 246, it is evident that a stay rod $1\frac{3}{8}$ ins. in diameter is indicated. Since a large portion of the area included in the above calculation lies near the stiff flanges of the tube sheet, a rod $1\frac{1}{4}$ ins. in diameter will be considered adequate. There is no reach of unstayed plate in this arrangement greater than 6.24 ins., half the diagonal of the square to which reference was made above. The stay rod passes to the front head at a slight angle and is anchored in an extension of the handhole pad. The staying having been satisfactorily designed, the tentative tube sheet thickness, $\frac{11}{16}$ in., will be accepted.

Riveted Joints. — There are three riveted joints demanding attention in this boiler, *viz.*: the longitudinal seam, the external ring seam and the furnace seam. The latter is exposed to the fire and hence its calking limit must be established by reference to the figures upon p. 132. The external seams are cool and therefore long pitches may be used.

In selecting a joint it is to be noted that an efficiency of at least 94 per cent is required. Joint *O*, the proportions of which are given in Table XXI, will be used therefore for the longitudinal seam. The external ring seams will be double-riveted lap joints, Table VIII, and the furnace seams single-riveted lap joints, Table VII.

The following data are necessary in ascertaining the pitch and laps for the various joints:

Thickness of shell	$\frac{5}{8}$ in.
Thickness of tube sheet	$\frac{11}{16}$ in.
Thickness of furnace tube	$\frac{7}{16}$ in.
Thickness of inside cover plate	$\frac{1}{2}$ in.
Thickness of outside cover plate to be governed by the necessities in regard to calking.	

With a rivet $1\frac{1}{16}$ ins. in diameter the pitch as taken from Table XXI is 19.39 ins. The calking pitch along the outer row will therefore be

$$\frac{19.39}{4} = 4.85 \text{ ins.}$$

Referring to Table XXIII, p. 173, it is noted that the limit of calking for a working pressure of 150 pounds with $1\frac{1}{16}$ in. rivet is 5.03 ins. and that with this figure an outer cover plate at least $\frac{9}{16}$ in. in thickness must be used. The joint will therefore be made up with such an outside cover plate since the thickness of the latter does not directly affect the value of the pitch. The efficiency of this joint is 94.5 per cent, a figure well in excess of that specified in the requirements.

The rivets in the outer rows of this joint fail by shearing and those upon the inner by crushing, hence the following lap values may be determined:

Inside cover plate.	Table III.	1.61 ins.	Use $1\frac{5}{8}$ ins.
Outside cover plate.	Fig. 76, $\frac{t}{t_1} = 1.1$	1.42 ins.	Use $1\frac{7}{16}$ ins.
Main plate.	Table IV.	1.75 ins.	Use $1\frac{3}{4}$ ins.

Starting at the center of the joint the first row of rivets is located by the lap value given above. The second row is placed $2\frac{1}{2}$ ins. from the first as found graphically by the method of Fig. 52, p. 101. To provide clearance for calking, the third row is driven at a distance of $1\frac{1}{2}$ ins. from the edge of the outside cover plate. To allow sufficient room for driving, the distance between the outer rows is made $2\frac{1}{2}$ ins.

The pitch for the double-riveted ring seams using rivets $1\frac{1}{16}$ in. in diameter as taken from Table VIII is 3.39 ins. and the corresponding efficiency 68.6 per cent. The distance between the staggered rows in this joint as found from formula (72), p. 100, is 1.98 ins. The rows will be spaced 2 ins. apart. The circumference of a circle 84 inches in diameter is 263.9 ins. Dividing the latter by the pitch indicates 77.8 spaces in the ring seam. To make this number a multiple of four the total number of ring seam pitches will be made 80, corresponding to a pitch value of 3.30 ins. and an efficiency of 67.9 per cent. The latter figure is calculated from equation (78), p. 114, since the pitch has been slightly reduced. The lap for the ring seam must be based upon the shearing of the rivets. The value from Table III, p. 137, for plate having a thickness of $\frac{5}{8}$ in. and rivets $1\frac{1}{16}$ ins. in diameter is 1.49 ins. A lap $1\frac{1}{2}$ ins. wide will be used.

The single-riveted lap joint *A*, Fig. 83, used for the furnace seams would generally employ a smaller diameter of rivet in order to place less metal in the region of the fire. Selecting therefore for this joint a rivet $\frac{15}{16}$ in. in diameter, the pitch from Table VII, p. 157, is 2.23 ins. This would give a net distance $p - d$ between rivet shanks of 1.29 ins. which is below the usual amount allowed in fire seams as indicated upon p. 132. The efficiency of this joint with the above theoretical pitch is 57.9 per cent. The circumference of a circle having a diameter equal to that of the outside of the furnace end is 138.6 ins. The pitch of this joint can safely be increased to

$$1.375 + 0.938 = 2.313 \text{ ins.}$$

Dividing the circumference of the furnace seam by the latter quantity it is evident that 59.9 spaces may be used. In order to make the total number of pitches divisible by four, 60 spaces will be used, the corresponding distance between rivet centers being 2.310 ins.

The efficiency of this joint having an arbitrarily increased pitch is dependent upon the shearing failure of the rivets.

Referring to formula (79), p. 115,

$$\begin{aligned} V &= \frac{(0.69)\,(45{,}000)}{(2.310)\,(\frac{7}{16})\,(55{,}000)} \\ &= 0.559 \text{ or } 55.9 \text{ per cent.} \end{aligned}$$

This efficiency is well in excess of that usually required for ring seams and hence is satisfactory.

Since the pitch has been arbitrarily increased in this seam, the lap will be based upon the single shearing of the rivets rather than tearing of the plate. Its value from Table III, p. 137, for plate $\frac{7}{16}$ in. in thickness with rivets $\frac{15}{16}$ in. in diameter, is 1.42 ins. A lap of $1\frac{7}{16}$ ins. will be used. Since it is desired to keep the metal in this joint to the lowest possible amount consistent with strength a different lap will be used in the tube sheet flange, based upon the thickness of the latter. By reference to the same table under $\frac{11}{16}$ in. plate and $\frac{15}{16}$ in. rivets, the lap value is found to be 1.23 ins. A lap of $1\frac{1}{4}$ ins. will be used.

Insertion of Longitudinal Joint. — The length of plate between the inner rows of rivets in the tube sheet ring seams, as determined upon the drawing, is 154.5 ins. From this must be deducted 2 ins., the distance between the staggered rows of the middle ring seam, leaving a net length of 152.5 ins. The pitches of the longitudinal joint must be arranged to fill this space without seriously altering the efficiency. Since the plate thickness is so great the form of joint end shown in Fig. 82, p. 144, will be used. The total number of small pitches will then be

$$\frac{152.5}{4.85} = 31.45.$$

Trying 32 pitches, the exact value of the latter will be

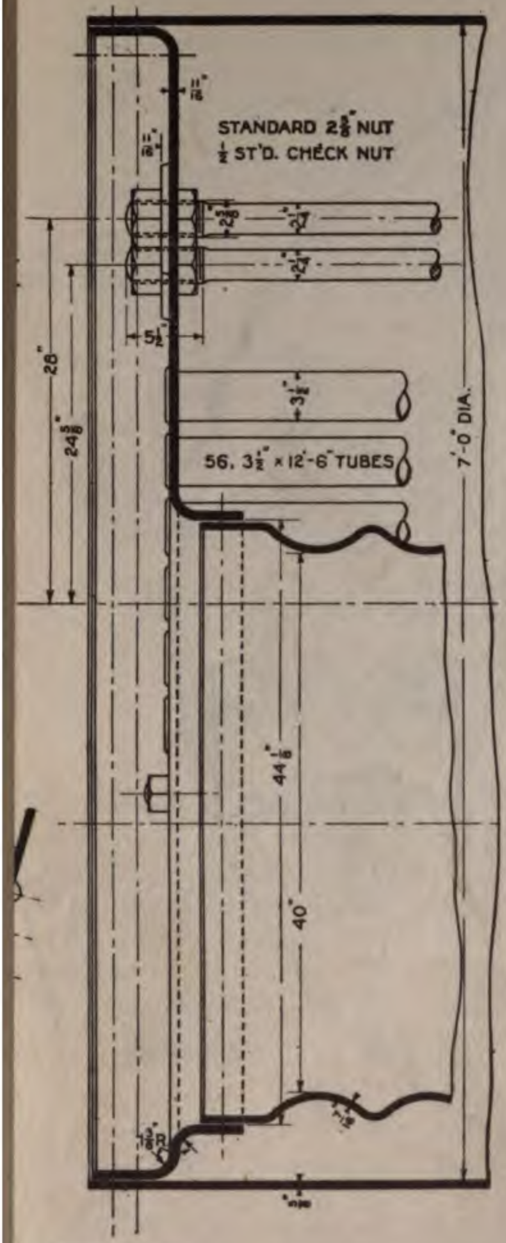
$$\frac{152.5}{32} = 4.765 \text{ ins.}$$

and the corresponding longitudinal pitch 19.06 ins. If the end spaces p_2 are not contracted the efficiency will be

$$\frac{19.06 - 1.06}{19.06} = 0.9445,$$

or 94.5 per cent, a figure somewhat in excess of the stipulated amount. By slightly displacing two inside rivets at the end of each joint as shown in the drawing, room for assembling and driving all the rivets will be secured. The two seams, each containing four long pitches, are therefore identical and the design may be considered satisfactory.

The drawing shows the attachment of the steam nozzles, manhole, feed and blow-off pipes, and water column connections called for in the specifications. The smoke bonnet is generally dependent for its exact arrangement upon the external conditions surrounding the boiler, hence it has not been shown. An accurate location of the fusible plug in the rear head requires the roof of the combustion chamber to be placed so high as to conflict somewhat with the stay rod nuts. The brick work may be


11/16"
11/16"
STANDARD 2 3/8" NUT
1/2 ST'D. CHECK NUT
2 3/8"
2 1/4"
2 1/4"
5 1/2"
28"
24 5/8"
3 1/2"
56, 3 1/2" × 12'-6" TUBES
7'-0" DIA.
44 1/8"
40"
1 3/16"R

BOILER

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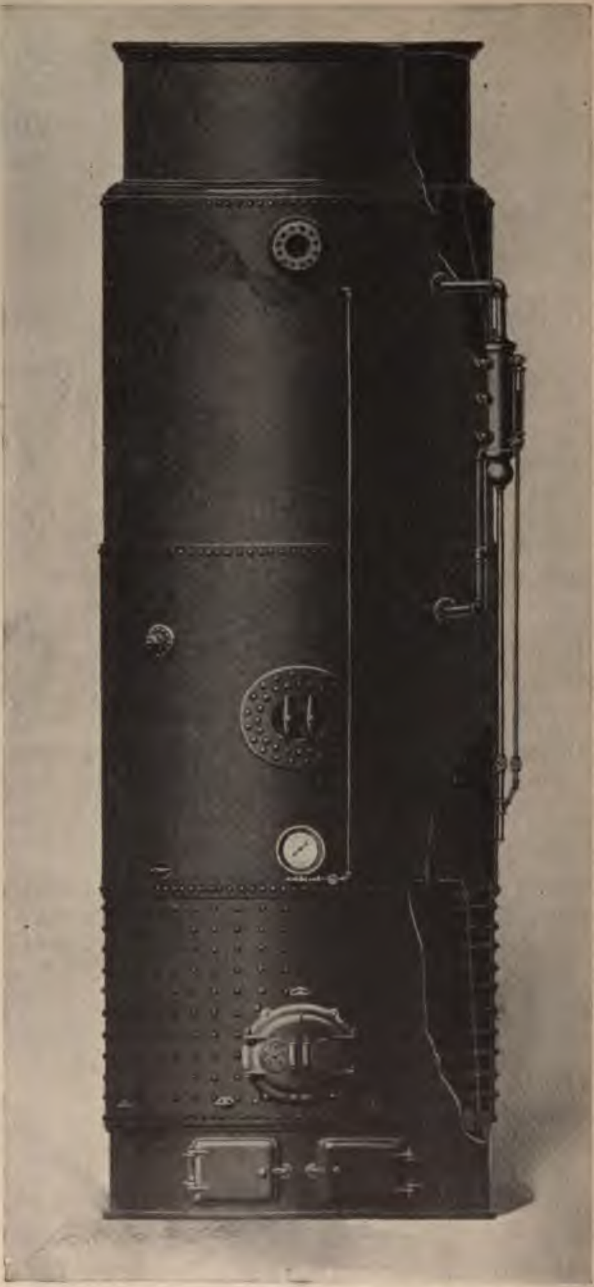
ASTOR, LENOX
TILDEN FOUNDATIONS

chipped slightly, however, to provide the necessary clearance room.

In accordance with the above calculations the tube sheet drawing, Fig. 188, and the working drawings, Fig. 189, have been prepared.



VERTICAL STRAIGHT SHELL BOILER.
International Engineering Works.

FIG. 190.

CHAPTER VII.

DESIGN OF A VERTICAL STRAIGHT SHELL MULTI-TUBULAR BOILER.

The specifications and complete design of a straight shell multi-tubular boiler, similar to the one described below, are given in this chapter.

A vertical straight shell tubular boiler is shown in Fig. 190. As its name signifies the external shell is cylindrical and straight throughout its length. This gives a larger steam space and less favorable conditions for superheating than in the other types of vertical boilers, *viz.*, the tapered course and Manning. The facilities for inspection, however, are much better than in the latter types. The external shell is in three courses. The upper courses are preferably identical, with multi-riveted butt-joints. They may be made of any desired thickness since they do not come in contact with the fire. The lower course, or external furnace sheet, is usually thinner than the other courses since the numerous stay bolts materially reinforce it. The joint is of the butt-type with outside cover plate only. The boiler being under no transverse strain, single-riveted lap joints suffice for ring seams throughout. The furnace is contained within the shell and consists of rather thin boiler plate with a single-riveted lap joint for its vertical seam. The bottoms of the internal and external furnace sheets are securely riveted to the mud-ring — a wrought iron or soft steel ring of rectangular cross section. The lower tube sheet, or crown sheet, is riveted to the top of the furnace. The upper tube sheet is of the ordinary type. Numerous tubes of rather small diameter arranged in straight lines give both water-heating and superheating surface and form a direct course for the flames from the grate to the uptake. To permit of inspection the tubes are kept back about one foot from the shell. This leaves the outer portion of the upper tube sheet unstayed and a circular row of diagonal braces is necessary to support it. A few tubes are usually omitted opposite the manhole and additional braces may be required to support the plate thus left unstayed. To avoid the use of stays in the lower tube sheet, the furnace sheet is tapered about two inches from the bottom to the top, thus bringing the stiff tube sheet flange fairly close to the tubes and making braces unnecessary. Numerous screwed stay bolts headed over cold on the inside and outside connect the furnace sheets, thus preventing the inner one from collapsing and the outer one from bulging. The grate is on the level of the mud-ring and is usually of the segmental type, although the larger boilers may use the sector pattern. Allowance must be made for expansion, hence the grate diameter is somewhat less than that of the bottom of the furnace.

The furnace rests on a cast iron ash-pit frame with double doors. A fire door is supported by a frame fastened to the lower course. A manhole in the middle course provides an entrance to the boiler for inspection. Two steam nozzles are placed near the top of the boiler on opposite sides, one for the steam pipe and one for a safety valve. Since sediment will collect on horizontal surfaces handholes are located to give access to such surfaces, *i.e.*, the mud-ring, the crown sheet and the flanging over the fire door. A feed pipe flange with bushing serves as a coupling between the external and internal feed pipes. Provision is also made for water column and blow-off connections. One extra heavy tube is inserted with a small fusible plug to give warning of low water and overheating. A smoke bonnet is riveted to a structural angle which in turn is riveted to the upper tube sheet.

76. Specifications. — The following general specifications apply to vertical boilers as a class:

Shell Plates. — Plates shall be of the best quality O. H. Firebox Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	52,000
Not more than	62,000
Yield point, min. lbs. per sq. in	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T. S.}}$

Heads. — Heads shall be of best O. H. Flange Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	55,000
Not more than	65,000
Yield point, min. lbs. per sq. in	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T.S.}}$

All heads are to be machine-flanged by the spinning process to an inner radius not less than twice the thickness, and thoroughly annealed.

Rivets. — Rivets shall be of the best quality of soft steel according to the following specifications:

Tensile strength, lbs. per sq. in.:	
Not less than	45,000
Not more than	55,000
Elongation in 8 ins., min. per cent, but need not exceed 30 per cent	$\frac{1,500,000}{\text{T.S.}}$
Shearing strength, lbs. per sq. in.	45,000
Crushing strength (bearing pressure), lbs. per sq. in. of projected area	96,000

Seams and Riveting. — The external vertical seams shall be of the butt-joint type with inside and outside cover plates. The circular or girth seams in the shell shall be of the single-riveted lap-joint type; the furnace seam shall be of the single-riveted lap-joint type. The shell seam outside of the furnace shall be of the double-riveted butt-joint type with outside cover plate only. The vertical seams shall break joints in the several courses.

Rivets shall be $\frac{1}{16}$ in. smaller in diameter than the holes they are to fill. All holes shall be drilled; *i.e.*, they shall be punched $\frac{1}{4}$ in. small and then drilled to size with all plates and cover plates in place, after the plates have been rolled. After drilling the holes the plates shall be taken apart and the burrs removed from the edges of the holes. All riveting shall be done with a hydraulic riveting machine wherever practicable, pressure being kept on each rivet until it has taken its shrinkage to insure tight joints. Rivet heads, on all machine-driven rivets, shall be of the type ordinarily used in pressure work. Where necessary, the rivets may be pneumatically driven, the formed head taking the above shape, the other retaining its original shape.

Rolling. — The plates shall be rolled cold by gradual and regular increments to the exact radius required and the whole circumference rolled to a true circle. Butt straps shall be rolled to the same radius as the shell in special forms made for that purpose.

Planing. — The edges of the plates shall be beveled to an angle of about 75 degrees on a plate planing machine. After the heads have been flanged and annealed, the calking edge shall be turned off on a mill to the same bevel as the shell plates.

Calking. — All seams shall be carefully calked with a pneumatic hammer, using a round-nosed calking tool.

Tubes. — Tubes shall be made of the best American lap-welded or seamless steel and arranged in straight rows at a uniform distance apart. To assist in cleaning, two wide avenues at right angles to one another shall be left through the tube arrangement. The tubes in general shall have at least one inch of clear water around them and the avenues shall be at least 3 ins. wide in the clear.

Tube holes shall be punched not over one inch in diameter and drilled to size with a revolving cutter, and the edges chamfered to a radius of $\frac{1}{16}$ in. Tube ends shall be carefully expanded and beaded over with pneumatic tools.

Staying of Upper Tube Sheet. — The upper head outside of the area occupied by the tubes shall be stayed and the arrangement shall be such as to secure an even spacing of the stayed points and a good support for the entire surface. The reaches of plate between stays in the tube sheet shall not exceed in length those allowed for stay bolts on flat plates of equal thickness. The stays shall be so spaced as to secure the most uniform loading possible. Braces shall be of an approved weldless steel type, securely riveted to shell and head and shall conform to the standard dimensions and specifications for such stays. The greatest allowable angle between braces and shell shall be 20 degrees.

Staying in Water Legs. — The sheets of the internal furnace shall be thoroughly stayed to the outer shell in a regular and systematic manner. The stay bolts shall be arranged in squares and the spacing shall not exceed that allowed on flat plates of equal thickness. The material and proportions of the stay bolts shall be those specified for standard solid boiler stay bolts. The seams in the inside and outside furnace sheets shall not be in close proximity to one another. A vertical row of stay bolts shall be located near the single-riveted lap joint in the inside furnace sheet. The stay bolts shall be securely headed over cold upon the inside and outside by a pneumatic riveter. The outside furnace sheet shall extend as high as the lower tube sheet. A slope of not over 2 ins. from mud-ring to lower tube sheet shall be permissible in the internal furnace sheet so as to avoid unstayed plate in the lower tube sheet outside of the tubes.

Mud-Ring. — The mud-ring shall be a welded continuous ring of rectangular cross-section shaped to an accurate circle. Holes for the rivets shall be drilled accurately and in a true radial direction.

Manhole. — One 11 ins. × 15 ins. manhole shall be located in the shell on the front of the boiler, in the center of the middle course. It shall have a pressed steel frame or ring and shall be fitted with a pressed steel cover, bolt, yoke and a copper or rubber gasket. The frame shall have a net cross-sectional area on a line through its center parallel to the axis of the shell, whose tensile resistance is at least equivalent to the tensile resistance of the shell plate removed on the same line.

The frame shall be riveted to the shell with two rows of rivets. The resistance of the rivets in shear or in crushing on one side of the longitudinal section through the center of the frame shall not be less than the tensile resistance of the plate removed on the same section.

Handholes. — Handholes shall be located near the bottom of the middle course so placed as to communicate with the avenues in the tubes. Handholes shall be located near the bottom of the external furnace sheet to facilitate cleaning the bottom of the water leg. These handholes shall be symmetrically spaced. One handhole shall be located directly above the fire door.

Nozzles. — Two cast steel nozzles of approved design shall be machine-riveted to the shell for main steam and safety-valve connections. The flanges which join the shell shall be beveled and calked on the outside. They shall be placed about 180 degrees from one another and located about 12 ins. below the top ring seam.

Feed Pipe. — About halfway between the manhole and ring seam next above, a feed pipe flange and bushing shall be located. The entrance shall be in line with one of the avenues in the tubes. An internal feed pipe screwed into the bushing from within shall pass to the center of the boiler. The size of pipe specified shall be used throughout.

Blow-off Connection. — A blow-off pipe of the specified size shall be located as low as possible over the mud-ring. A reinforcing pad of the same thickness as the external furnace sheet shall be securely riveted to the latter. In general, several of the mud-ring seam rivets and one or two stay bolts are used in attaching the reinforcement.

Water Column Connections. — In the side of the upper courses, tappings of the size specified for the water column connections shall be located about 7 ft. apart vertically, and so arranged as to readily accommodate the mean water level. To reinforce the holes either pads of boiler plate or standard flanges may be used.

Fusible Plug. — In a place conveniently accessible an extra heavy tube shall be inserted. A fusible plug shall be screwed into this tube from the outside with its center located at least one-third the tube length above the lower tube sheet or crown sheet. Room shall be provided to use a small wrench in screwing home the plug. The fusible core shall be so tapered as to withstand the pressure as it is applied in this case.

Smoke Bonnet. — With as large a diameter as possible a 2 ins. × 2 ins. × $\frac{5}{16}$ in. angle iron shall be riveted to the top tube sheet. To this angle a sheet iron uptake shall be riveted having an opening through the top or side, the area of which is about 90 per cent of the internal transverse tube area.

Fire Door Opening. — The internal and external furnace sheets shall be flanged out a sufficient distance to bring the fire door seam within reach of the riveting machine. The opening shall be elliptical in shape and shall measure 19 ins. × 24 ins. in the clear. Four tapped holes shall be provided to attach the door frame.

Ash-Pit. — A cast iron ash-pit frame shall be provided with double doors. The top where the mud-ring rests shall be narrow enough so that leakage in either the internal or external mud-ring seam may be calked tight without lifting the boiler.

77. Statement of Problem. — The calculations necessary in connection with the design of a vertical straight shell boiler follow:

Design a vertical straight shell multitubular boiler to conform to the preceding specifications and to embody the following general dimensions:

General Dimensions:

Rated Horse-Power (A.S.M.E. standard)		150
Working Pressure, lbs. per sq. in		150
Length of Tubes, ft		15
Number of Courses		3
Diameter of Steam Nozzles (2), ins		5
Handholes	Number	Size, ins.
Mud-ring,	6	$2\frac{5}{8} \times 3\frac{3}{4}$
Crown sheet,	4	4×6
Fire door,	1	$2\frac{5}{8} \times 3\frac{3}{4}$
Diameter of Feed Pipe, ins		$1\frac{1}{2}$
Diameter of Blow-off Pipe, ins		2
Diameter of Water Column Connections, ins		$1\frac{1}{4}$

Type of Staying for upper Tube Sheet	Diagonal
Kind of Coal	Bituminous

Types of Joint to be used:

Ring Seam Joints	Single-Riveted Lap.
Vertical Joints: Two Upper Courses	Butt-joint with inside and outside cover plates, having an efficiency of at least 93 per cent.
Outside Furnace Sheet	Butt-joint with outside cover plate only.
Inside Furnace Sheet	Single-Riveted Lap.

Design to include:

(*a*) Complete calculations.

(*b*) Working drawings, fully dimensioned, of

(1) Tube sheets. Plan drawing showing half view of upper and half of lower tube sheet, including staying and tube arrangement. Scale 3 ins. = 1 ft.

(2) Boiler. Half plan, half inverted plan, front elevation and development of joints. Scale 1½ ins. = 1 ft.

The following specific values of ratios and constants, for which a general discussion is given in Chap. IV, will be assumed:

Rate of Evaporation, lbs. of water per lb. of coal, about	9.5
Rate of Combustion, lbs. per sq. ft. of Grate Surface, per hr., about	18.5
Ratio: Length to Outside Diam. of Tubes, about	72
Ratio: Internal Transverse Tube Area to Grate Area, about	1 : 5.5
Steam Space per Boiler H.P., cu. ft., at least	0.80
Ratio: Steam Space to Water Space, about	1 : 3
Total Heating Surface per Boiler H.P., sq. ft., not less than	12
Ratio: Water and Superheating Surface to Grate Surface, about	60
Ratio: Water Heating Surface to Grate Surface, about	45
Ratio: Superheating Surface to Water Heating Surface	1 : 3
Reach of Riveting Machine, ft	8
Grate Diameters vary by inches.	
Least Factor of Safety	5
Height of Furnace, about ¾ Diameter of Grate.	
Width of Space outside of Tubes for inspection, ins., about	12
Grate Surface per H.P., sq. ft., about	0.20

78. Calculations.

Grate. — From the given horse-power and the standard A.S.M.E rating, p. 5, the water evaporated per hour is found to be

$$150 \times 34.5 = 5175 \text{ lbs. per hour.}$$

With the given rate of evaporation this will correspond to

$$\frac{5175}{9.5} = 544.7 \text{ lbs. of coal per hour.}$$

Since each square foot of grate burns 18.5 lbs. of coal per hour, the theoretical grate area will be

$$\frac{544.7}{18.5} = 29.4 \text{ sq. ft.}$$

The diameter corresponding is 73.5 ins.

Use grate diameter, 74 ins. Area 29.88 sq. ft.

Furnace Dimensions. — With one-half inch clearance, the inside diameter of the furnace at the mud-ring is 75 ins., and at the top, 71 ins., allowing a 2 in. taper for the furnace. The furnace height is $\frac{3}{4} \times 74 = 55\frac{1}{2}$ ins. Use 56 ins.

Diameter of Boiler. — The thickness of the inside furnace sheet may be taken as $\frac{7}{16}$ in., and the outside furnace sheet as $\frac{9}{16}$ in. Then the boiler diameter will be

$$75 + \tfrac{7}{8} + 6 + \tfrac{9}{8} = 83 \text{ ins.,}$$

allowing for a mud-ring 3 ins. wide.

Tubes. — The diameter of the tubes, from the ratio given, is

$$\frac{15 \times 12}{70} = 2.57 \text{ ins.}$$

Tubes $2\frac{1}{2}$ ins. in diameter will be used.

From Table XXIX, p. 190:

Internal circumference	= 7.17 ins.,
External transverse area	= 4.91 sq. ins.,
Internal transverse area	= 4.09 sq. ins.

Internal transverse tube area, from the ratio given, is

$$\frac{\text{Grate Area}}{5.5} = \frac{29.88}{5.5}$$
$$= 5.44 \text{ sq. ft. or } 783 \text{ sq. ins.}$$

The number of tubes required to give this transverse area will be

$$\frac{783}{4.09} = 191.3.$$

Use 191 tubes as a trial number in the tube sheet lay-out.

Water Levels. — With the data already found it is desired to determine how great a discrepancy there may be between the water levels as obtained from the ratio of steam space to water space (normal mean water level) and the water level which will secure the required steam space per horse-power (maximum mean water level). The relation of these water levels will give the designer an idea as to necessary changes. Should they come reasonably close, it remains to determine whether or not the other ratios depending upon water level are satisfied, and which level will satisfy them best as a whole.

The total volume of the boiler is

$$(\text{Tube length} + \text{height of furnace})\ (\text{Area 83 in. circle})$$

$$(15 \times 12 + 56)\ (5411) = 1{,}277{,}000 \text{ cu. ins. or } 739.1 \text{ cu. ft.}$$

The furnace volume is

$$(\text{Height of furnace})\ (\text{Area 73 in. circle})$$

$$56 \times 4185 = 234{,}400 \text{ cu. ins. or } 135.5 \text{ cu. ft.}$$

The volume of 191 tubes per foot of length is

$$191 \times 4.91 \times 12 = 11{,}254 \text{ cu. ins. or } 6.51 \text{ cu. ft.}$$

The total volume of tubes is

$$6.51 \times 15 = 97.65 \text{ cu. ft.}$$

Subtracting the volume of tubes and furnace from the total volume we find the steam space plus the water space to be

$$739.1 - 135.5 - 97.7 = 505.9 \text{ cu. ft.}$$

The steam space is to be one-fourth of this or

$$\frac{505.9}{4} = 126.5 \text{ cu. ft.}$$

Let the height of steam space necessary to give this volume be x ft., then

$$\frac{5411\,x}{144} - 6.51\,x = 126.5,$$

$$31.07\,x = 126.5,$$

$$x = 4.07 \text{ ft. or } 48.84 \text{ ins.}$$

which is the distance of the normal mean water level from the top tube sheet.

Allowing 0.80 cu. ft. of steam space per horse-power, the steam space should be

$$150 \times 0.80 = 120 \text{ cu. ft.}$$

To find the height x necessary to give this volume note that the volume of the boiler per foot of length, exclusive of the tubes, is

$$\frac{5411}{144} - 6.51 = 31.07 \text{ cu. ft.}$$

Then

$$31.07x = 120,$$
$$x = 3.86 \text{ ft. or } 46.32 \text{ ins.}$$

which is the distance of the maximum mean water level from the top tube sheet.

As the amount of water heating surface in this type of boiler is rarely as much as might be desired and the superheating effect is of minor importance, the higher of the two water levels will be selected.

Trial Heating Surface. — The heating surface of the furnace, based on a mean diameter of 73 ins. is

$$229.3 \times 56 = 12,840 \text{ sq. ins.}$$

The water heating surface of the tubes is

$$191 \times 7.17\,(180 - 46.32) = 183,100 \text{ sq. ins.}$$

The heating surface of the lower tube sheet is

Area 71 ins. circle − external transverse area of tubes

$$3959 - 191 \times 4.91 = 3021 \text{ sq. ins.}$$

Total water heating surface is

$$12,840 + 183,100 + 3021 = 198,961 \text{ sq. ins.}$$

The superheating surface of the tubes is

$$191 \times 46.32 \times 7.17 = 63,400 \text{ sq. ins.}$$

The superheating surface of the upper tube sheet is

Area 83 ins. circle − external transverse area of tubes

$$5411 - 191 \times 4.91 = 4474 \text{ sq. ins.}$$

The total superheating surface is now known to be

$$63,400 + 4474 = 67,874 \text{ sq. ins.}$$

The several ratios may now be determined to ascertain if any radical change in design is necessary.

Total heating surface per horse-power is

$$\frac{198,961 + 67,874}{144 \times 150} = 12.35 \text{ sq. ft.}$$

which is a satisfactory figure.

The ratio of superheating surface to water heating surface, desired as 1 to 3, is

$$\frac{67,874}{198,961} = \frac{1}{2.93},$$

which is a reasonably close approximation.

Thickness of Shell, Tube Sheets and Furnace Sheet. — With a joint efficiency of 93 per cent the thickness of shell is found from the plot, Fig. 11, p. 57, by interpolation to be 0.61 in.

The shell thickness will be taken $\frac{5}{8}$ in., the furnace sheets, as already mentioned, will be $\frac{7}{16}$ in. and $\frac{9}{16}$ in. respectively. The thickness of the crown sheet will be made $\frac{7}{16}$ in. and the upper tube sheet $\frac{9}{16}$ in.

Tube Sheet Lay-out. — The arrangement of tubes is now determined in accordance with the principles laid down in Art. 66, p. 228. It is found that 197 tubes may be used to advantage. The six additional tubes will not materially affect any ratios previously found. They will, indeed, tend to increase the heating surface which is desirable.

Final Ratios. — The final ratios called for on the tube sheet drawing may now be computed and compared with those given in the problem.

Grate Area, G. — The grate area will be 4301 sq. ins. or 29.88 sq. ft. as already determined.

Internal Transverse Tube Area, C. — The internal transverse tube area amounts to 805.7 sq. ins. or 5.60 sq. ft., this being the cross section of 197 tubes, at 4.09 ins. per tube.

Heating Surface, H. — The heating surface is of two kinds, *viz.*, water heating surface, *W.H.S.*, and steam heating surface or superheating surface, *S.H.S.*

The water heating surface will comprise, (*a*) the inside surface of the tubes to the mean water level, (*b*) the furnace sheet, (*c*) the crown sheet, exclusive of tube sheet holes. These several quantities are:

$$\begin{array}{lrl}
(a)\ 197 \times 7.17\,(180 - 46.3) & = & 188{,}850 \text{ sq. ins.} \\
(b)\ 229.3 \times 56 & = & 12{,}841 \text{ sq. ins.} \\
(c)\ 3959 - 197 \times 4.91 & = & 2{,}992 \text{ sq. ins.} \\
W.H.S. & = & \overline{204{,}683} \text{ sq. ins. or } 1421.4 \text{ sq. ft.}
\end{array}$$

The superheating surface consists of the remaining tube area and the top tube sheet exclusive of the tube sheet holes, or

$$\begin{array}{lrl}
197 \times 7.17 \times 46.3 & = & 65{,}400 \text{ sq. ins.} \\
5411 - 197 \times 4.91 & = & 4{,}444 \text{ sq. ins.} \\
S.H.S. & = & \overline{69{,}844} \text{ sq. ins. or } 485 \text{ sq. ft.} \\
\text{Hence } H = 1421.4 + 485 & = & 1906.4 \text{ sq. ft.}
\end{array}$$

Steam Space, S. — The steam space will be the volume of the boiler above the mean water level minus the tube volume in this space. This is readily obtained by multiplying the superheating surface of the top tube sheet, already found, by the distance down to the mean water level.

$$\begin{aligned} S &= 4444 \times 46.3 \\ &= 205{,}760 \text{ cu. ins. or } 119.1 \text{ cu. ft.} \end{aligned}$$

Water Space, W. — The water space includes the entire volume of the boiler, minus that occupied by the tubes, furnace and steam space.

$$\begin{aligned} \text{Vol. of boiler} &= (180 + 56)(5411) \\ &= 1{,}277{,}000 \text{ cu. ins.} \end{aligned}$$

$$\text{Vol. of furnace} = \text{Mean area} \times \text{Height}$$

or

$$4185 \times 56 = 234{,}400 \text{ cu. ins.}$$

Vol. of tubes is

$$197 \times 4.91 \times 180 = 174{,}100 \text{ cu. ins.}$$

$$\text{Steam space} = 205{,}760 \text{ cu. ins.}$$

Hence

$$\begin{aligned} W &= 1{,}277{,}000 - 234{,}400 - 174{,}100 - 205{,}760 \\ &= 662{,}740 \text{ cu. ins. or } 383.5 \text{ cu. ft.} \end{aligned}$$

The several ratios may now be obtained and compared with those desired.

$$(1) \qquad \frac{H}{G} = \frac{1906.4}{29.88}$$

$$= 63.8 \text{ (actual)} \quad \ldots\ldots \quad 60 \text{ (desired).}$$

$$(2) \qquad \frac{W.H.S.}{G} = \frac{1421.4}{29.88}$$

$$= 47.6 \text{ (actual)} \quad \ldots\ldots \quad 45 \text{ (desired).}$$

$$(3) \qquad \frac{S.H.S.}{W.H.S.} = \frac{485}{1421.4}$$

$$= \frac{1}{2.94}.$$

$$(4) \qquad \frac{S}{W} = \frac{119.1}{383.5}$$

$$= \frac{1}{3.11} \text{ (actual)} \quad \ldots\ldots \quad \tfrac{1}{3} \text{ (desired).}$$

$$(5) \qquad \frac{S}{H.P.} = \frac{119.1}{150}$$

$$= 0.795 \text{ (actual)} \quad \ldots\ldots \quad 0.80 \text{ (desired).}$$

(6)
$$\frac{C}{G} = \frac{5.60}{29.88}$$
$$= \frac{1}{5.44} \text{ (actual)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \frac{1}{5.5} \text{ (desired)}.$$

(7)
$$\frac{H}{H.P.} = \frac{1906.4}{150}$$
$$= 12.7 \text{ (actual)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad 12+ \text{ (desired)}.$$

(8)
$$\frac{G}{H.P.} = \frac{29.88}{150}$$
$$= 0.20 \text{ (actual)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad 0.20 \text{ (desired)}.$$

Staying. — The centers of the diagonal braces, Scully, Fig. 140, p. 250, for the upper tube sheet are placed halfway between the tube stiff line, a line approximately one-half the tube radius from the centers of the outside tubes, and the flange stiff line. From the drawing the radius of the stay rivet circle is found to be $34\frac{1}{16}$ ins. The width of the space to be supported by stays is $5\frac{1}{8}$ ins.

The diameter of the rivets used in the braces is $\frac{7}{8}$ in. and the upper tube sheet is $\frac{9}{16}$ in. thick. The following quantities should be noted for this rivet diameter and plate thickness, at the given working pressure:

Maximum spacing of stayed points as regards plate, Table XXXIX, p. 233, 7.22 ins.

Area that one rivet will support, Table XL, p. 235, 24.05 sq. ins.

The area which may be allotted to the two $\frac{7}{8}$ in. rivets of one brace is 48.10 sq. ins.

The length of area $5\frac{1}{8}$ ins. wide which the two rivets will support is then

$$\frac{48.10}{5.125} = 9.38 \text{ ins.}$$

To find the length of area which one stay can safely support, assume the angle of the stay with tube sheet to be 80 degrees, and a factor of safety of 8 on an ultimate tensile strength of 55,000 lbs. per sq. in. One stay will then support

$$\frac{55{,}000 \times 0.994 \times 0.9848}{8 \times 150} = 44.8 \text{ sq. ins.}$$

The length of area supported by one stay is

$$\frac{44.8}{5.125} = 8.76 \text{ ins.}$$

This, being less than that given for the rivets, will determine the spacing of braces. It is to be noted that in no place is there a length of plate between supported points greater than that allowed, *viz.*, 7.22 ins.

The number of braces is now readily obtained

$$\frac{\text{Circumference of } 68\tfrac{1}{8} \text{ in. circle}}{8.76} = 24.45.$$

Twenty-four stays will be used, six in each quarter of the tube sheet. This will reduce the factor of safety slightly, or from 8 to 7.85.

Stay Bolts in Water Leg. — Assuming stay bolts $1\tfrac{3}{16}$ ins. in diameter the allowable load per stay bolt, as determined from Table XLVI, p. 264, is 5718 lbs. The area which one stay bolt will support is

$$\frac{5718}{150} = 38.12 \text{ sq. ins.}$$

The maximum distance between stay bolts as regards plate is found to be 5.62 ins., Table XXXIX, p. 233, $\tfrac{7}{16}$ in. plate and 150 lbs. per sq. in. pressure. The area supported by one stay bolt would be

$$5.62 \times 5.62 = 31.60 \text{ sq. ins.}$$

This quantity being less than that previously found will determine the pitching of the stay bolts.

The number of spaces between ring seams will be

$$\frac{56}{5.62} = 9.96. \qquad \text{Use 10 spaces.}$$

The number of rows circumferentially will be, using the larger radius of the furnace,

$$\frac{75\,\pi}{5.62} = 41.8. \qquad \text{Use 42 spaces.}$$

Thus the pitch is

$$\frac{75\,\pi}{42} = 5.609 \text{ ins.}$$

The circumferential pitch on the outside furnace sheet is

$$\frac{83\,\pi}{42} = 6.207 \text{ ins.}$$

This is well within the 7.22 ins. allowed as per Table XXXIX, p. 233.

Design of Longitudinal Riveted Joint. — The type of butt-joint selected must be such that an efficiency of 93 per cent may be obtained with a main plate thickness of $\frac{5}{8}$ in. These conditions are satisfied by either of the quadruple-riveted butt-joints shown in Tables XXI and XXII. Joint *P*, page 172, will be selected. Assuming a rivet 1 in. in diameter, reference to Table XXII shows the maximum pitch to be 14.55 ins. with an efficiency of 93.1 per cent. The calking pitch p_1 is one-third of the calculated pitch or 4.85 ins. With an outside cover plate $\frac{1}{2}$ in. thick, this pitch will not calk. Hence it is necessary to increase the outside cover plate thickness to $\frac{9}{16}$ in., for which the calking pitch is 4.97 ins., Table XXIII, p. 173, which is satisfactory. This change does not affect the pitch. The pitch as determined from Table XXII is not independent of the inside cover plate thickness. Therefore, the latter must be taken $\frac{1}{2}$ in. as given in the table. The several laps on the vertical joints of the shell may now be found from the tables of Art. 48.

Outside cover plate, main joint.	Fig. 76, p. 139.	Lap = 1.35 ins.	Use $1\frac{7}{16}$ ins.
Inside cover plate, main joint.	Table III.	Lap = 1.48 "	Use $1\frac{1}{2}$ "
Shell, main joint.	Table IV.	Lap = 1.65 "	Use $1\frac{11}{16}$ "
Cover plate, outside furnace sheet.	Table III.	Lap = 1.55 "	Use $1\frac{9}{16}$ "
Outside furnace sheet.	Table III.	Lap = 1.43 "	Use $1\frac{7}{16}$ "

It should be noted that the laps of the single-riveted lap joints cannot be definitely determined until the pitches of these joints are known. Therefore the design of these joints will precede that of the longitudinal joint, the insertion of the latter being dependent upon the ring seam laps of the shell and furnace sheet.

Ring Seam Joint. — In determining the pitch for the ring seam rivets it should be noted that with the exception of the middle joint the two plates are of different thicknesses. From Table VII, p. 157, it is found that for 1 in. rivets in $\frac{5}{8}$ in. plate the maximum pitch is so small that the rivets cannot be driven. The same table shows that for $\frac{9}{16}$ in. plate the maximum pitch is 2.14 ins. with an efficiency of 53.3 per cent. Since it is desirable to have the same number of rivets in all of the girth joints, the above pitch will be assumed as a trial ring seam pitch. Its final acceptance will be dependent upon the efficiency resulting from its use in the middle joint where both plates are $\frac{5}{8}$ in. thick. The number of pitches in a ring seam will be

$$\frac{83\,\pi}{2.14} = 121.8.$$

Reducing this to 120, a number divisible by both 4 and 6, the resulting pitch is

$$\frac{83\,\pi}{120} = 2.17 \text{ ins.}$$

The efficiency of the middle ring seam may now be found from equation (79), p. 115, since the pitch it is proposed to use is greater than the theoretical maximum pitch.

$$\begin{aligned} V &= \frac{0.7854 \times 45{,}000}{2.17 \times 0.625 \times 55{,}000} \\ &= 0.4737 \text{ or } 47.37 \text{ per cent.} \end{aligned}$$

This figure is more than one-half of the longitudinal joint efficiency, and the above pitch may, therefore, be considered satisfactory.

The ring seam laps for the main plate, upper tube sheet and outside furnace sheet may now be found from Table III, p. 137.

Main plate	Lap = 1.38 ins.	Use $1\frac{3}{8}$ ins.
Upper tube sheet	Lap = 1.43 "	Use $1\frac{7}{16}$ "
Outside furnace sheet	Lap = 1.43 "	Use $1\frac{7}{16}$ "

Inside Furnace Sheet Joints. — The pitch of the rivets for the crown sheet, as determined from Table VII, p. 157, is 2.47 ins. As this is more than the $2\frac{3}{8}$ ins. allowed for fire seams, p. 132, the pitch must be reduced to the latter figure. Using $2\frac{3}{8}$ ins. as a limit the number of furnace seam pitches is

$$\frac{71\,\pi}{2.375} = 93.9.$$

Using 96, in order that the number may be divisible by 4, the pitch will be

$$\frac{71\,\pi}{96} = 2.33 \text{ ins.}$$

and the efficiency from formula (78), p. 114,

$$\begin{aligned} V &= \frac{2.33 - 1}{2.33} \\ &= 0.571 \text{ or } 57.1 \text{ per cent.} \end{aligned}$$

The maximum pitch for the rivets in the vertical lap joint on the furnace sheet is 2.47 ins. As this is also a fire seam its pitch must be reduced to approximately $2\frac{3}{8}$ ins. The distance between ring seams being 56 ins. the number of spaces called for will be

$$\frac{56}{2.375} = 23.58.$$

As the plates in this case are without curvature this number may be safely reduced to 23. The corresponding pitch is

$$\frac{56}{23} = 2.435 \text{ ins.}$$

The pitches in both of the above joints having been reduced the laps are obtained from the plot, Fig. 77, p. 140.

Crown sheet	Lap = 1.50 ins.	Use $1\frac{9}{16}$ ins.
Furnace sheet, ring seam	Lap = 1.50 "	Use $1\frac{9}{16}$ "
Furnace sheet, vertical joint	Lap = 1.53 "	Use $1\frac{9}{16}$ "

Insertion of Longitudinal Joint. — The distance between ring seams may now be found. Allowing $\frac{1}{2}$ in. for beading, the outside surfaces of tube sheets are 179.5 ins. apart. It is desired to place the crown sheet rivets and the upper ring seam for the outside furnace sheet at the same level. From the top of the upper tube sheet to the highest ring seam will be:

Outside radius of flange + $\frac{1}{16}$ in. (to insure flat plate for calking) + the shell lap, or

$$3 \times \frac{9}{16} + \frac{1}{16} + 1\frac{3}{8} = 3\frac{1}{8} \text{ ins.}$$

From the lower side of the crown sheet to its ring seam will be:

Outside radius of flange (taken 3 ins.) plus inside furnace sheet lap, minus thickness of crown sheet, or

$$3 + 1\frac{9}{16} - \frac{7}{16} = 4\frac{1}{8} \text{ ins.}$$

The distance between ring seams will be

$$179.5 - 3\frac{1}{8} + 4\frac{1}{8} = 180.5 \text{ ins.}$$

Allowing for 4 end pitches p_2, Fig. 98, ($0.8 \times 4.85 = 3\frac{7}{8}$ ins.) the remaining distance between the seams is

$$180.5 - 4 \times 3.875 = 165 \text{ ins.}$$

This will give

$$\frac{165}{4.85} = 34.05 \text{ short pitches } p_1.$$

Try 35 short pitches; on the upper course 16, and on the mi course 19, both of which are multiples of three, plus one.

The corresponding value of the short pitch is then

$$\frac{165}{35} = 4.714 \text{ ins.}$$

which is within the 3 per cent allowed, Art. 50, p. 142, witl changing the efficiency of the joint.

The width of outside cover plate is known when the dist between rivet rows is found. This may be obtained graphic Fig. 52, p. 101, or by use of equation (72), p. 100. It is $2\frac{7}{16}$ The outside cover plate width is

$$2\left(1\tfrac{7}{16} + 2\tfrac{7}{16} + 1\tfrac{11}{16}\right) = 11\tfrac{1}{8} \text{ ins.}$$

The distance between the outside rivet rows must be at l enough to allow $\frac{3}{16}$ in. clearance between heads, or $2\frac{3}{16}$ ins. taken $2\frac{1}{2}$ ins. The width of inside cover plate is then

$$2\left(1\tfrac{1}{2} + 2\tfrac{1}{2} + 1 + \tfrac{3}{16} + 1\tfrac{7}{16} + 2\tfrac{7}{16} + 1\tfrac{11}{16}\right) = 21.5 \text{ ins.}$$

With the joint satisfactorily inserted the final factor of sa may be obtained from equation (89), p. 117.

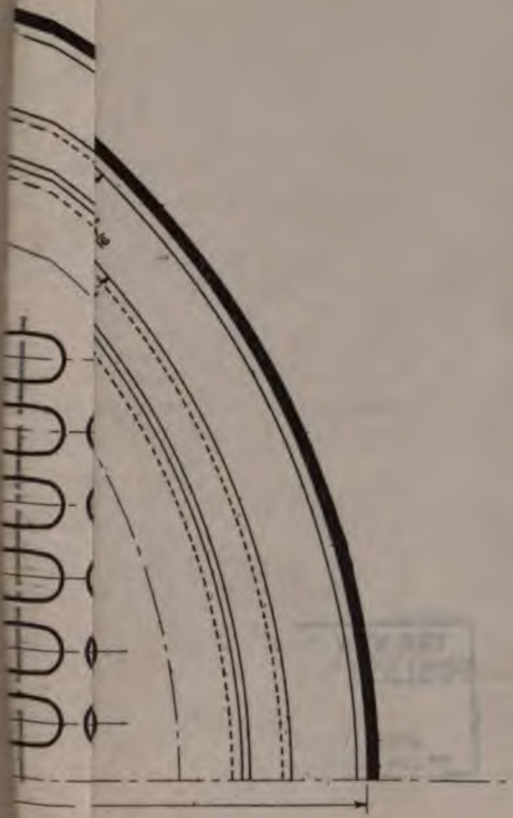
$$\text{F.S.} = \left(\frac{4.714 \times 3 - 1}{4.714 \times 3}\right)\left(\frac{55{,}000 \times 0.625}{150 \times 41.5}\right)$$

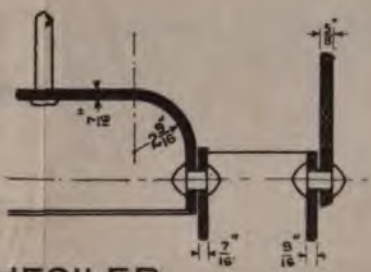
$$= 5.13.$$

Outside Furnace Sheet Joint. — The type of joint selecte that shown in Fig. 101, p. 149. Hence the pitch of the ri is one-half that of the stay bolts or 2.805 ins. Referenc Table XXIII, p. 173, shows this pitch to be well within calking limit. The distance between rows as determined f equation (72) or found graphically by the method of Fig. p. 101, is $1\frac{3}{4}$ ins. The half-width of cover plate is

$$1\tfrac{9}{16} + 1\tfrac{3}{4} + 1\tfrac{7}{16} = 4\tfrac{3}{4} \text{ ins.}$$

Riveting around Manhole and Steam Nozzle Openings. — arrangement of rivets for attaching the manhole ring is that gi in Fig. 156, p. 276. Two steam nozzles of specified size for ste pipe and safety valve connections are located 12 ins. from top ring seam. Rivets $\frac{7}{8}$ in. in diameter are used in b cases.





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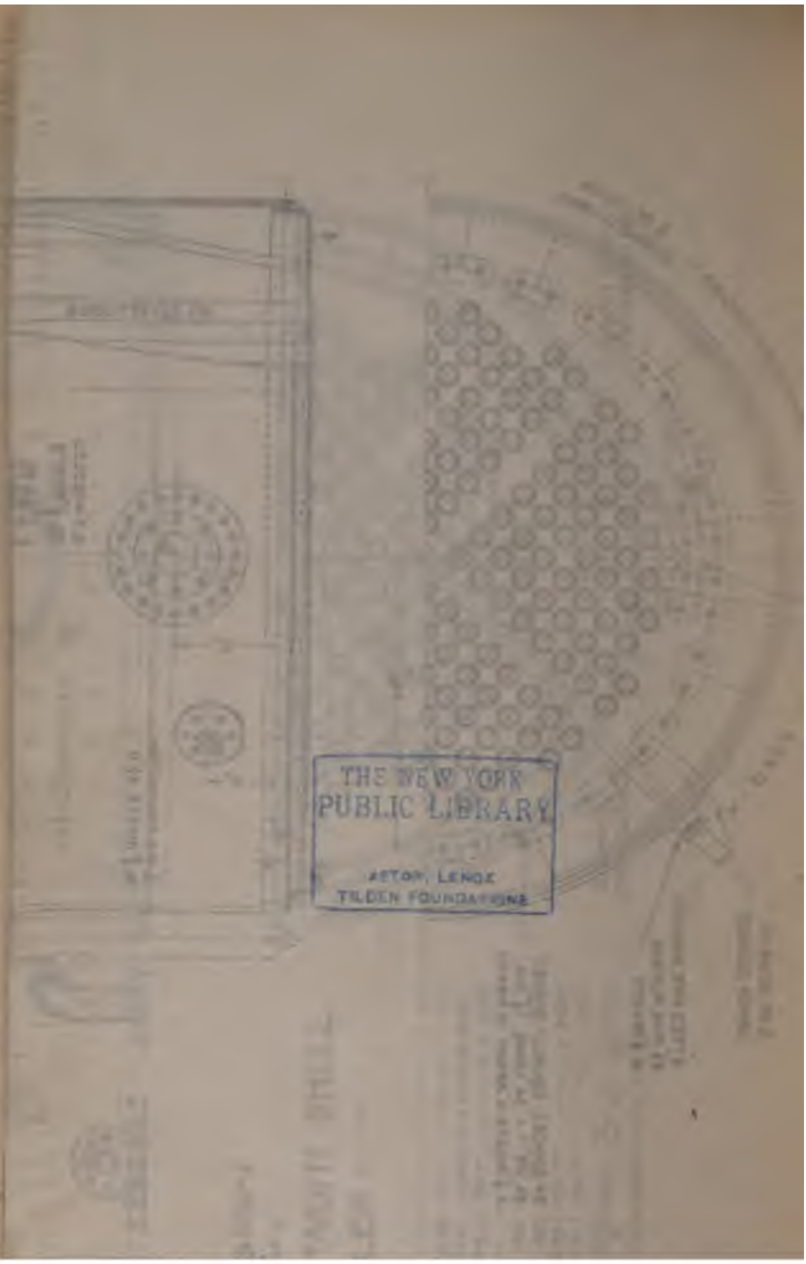

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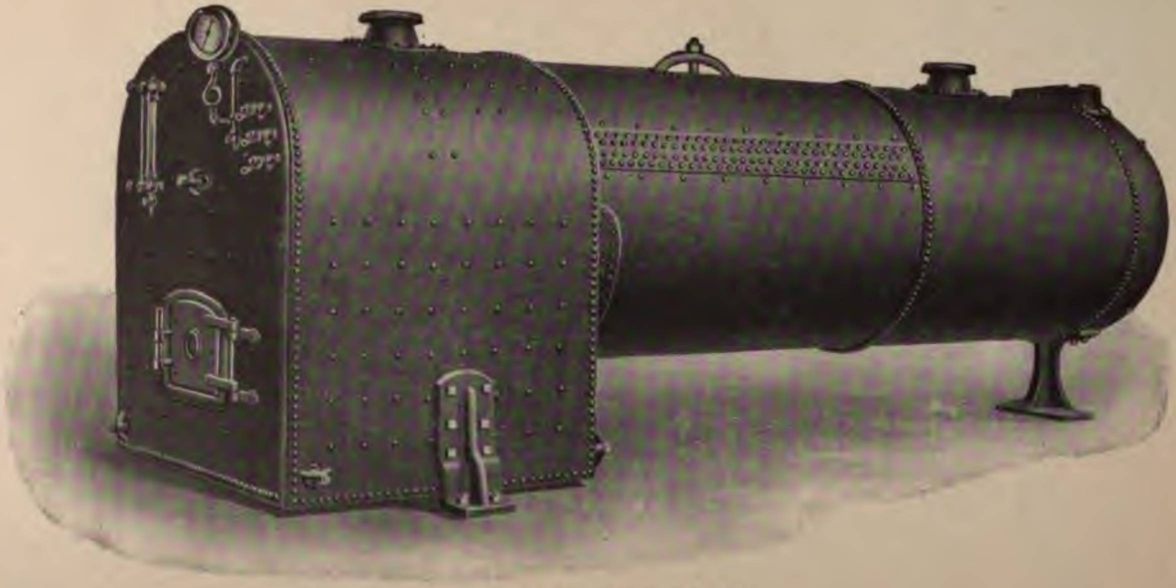
Water Column Connections. — Flanges for 1¼ in. pipe, as given in Table XLVII, p. 283, are located to receive the water column connections.

Feed Pipe Bushing. — The feed pipe bushing located in the rear of the middle course is that shown in Fig. 170, p. 288, for 1½ in. pipe.

The following drawings, Figs. 191 and 192, have been prepared to accompany the foregoing calculations.



LOCOMOTIVE TYPE BOILER
For Contractors' Use.
The Hodge Boiler Works.
FIG. 193.

CHAPTER VIII.

DESIGN OF A LOCOMOTIVE TYPE BOILER

For Contractors' Use.

In this chapter there will be worked out the complete design of a locomotive type boiler mounted on skids to illustrate the principles laid down in the previous chapters.

The illustration, Fig. 193, shows the usual form in which this boiler is manufactured. Its chief advantage is portability but when applied to a permanent installation and covered with insulating material it is also a very effective steam generator. The fire box consists of a rectangular chamber with either a flat or slightly curved roof. Water legs surround the furnace upon all sides and a copious layer of water is required above the crown sheet to prevent overheating. The front wall of the furnace constitutes the rear tube sheet. From the latter numerous fire tubes run to the front tube sheet. The tubes are spaced much after the manner of a horizontal cylindrical boiler. The exterior sides of the furnace course are vertical with a semi-circular portion at the top. Numerous stay bolts support the flat plate forming the water legs. A flanged fire door opening provides entrance for the fuel. Above the furnace the flat vertical plate at the rear of the boiler is stayed by diagonal braces running forward and anchored to the shell. The furnace roof is stayed either by crown bars or radial stays, depending upon the size and type of the boiler.

The outside wall of the water leg at the front end of the grate is called the throat sheet. Its upper edge is semi-circular in form and is flanged forward to conform to the curvature of the boiler shell forming the second course. The throat sheet and upper half of the furnace course complete the circle into which the middle course is riveted. The extension of the front course forms a smoke box from which an outlet leads upward.

Steam and safety valve nozzles are attached to the front and rear courses respectively. A manhole is installed in the center of the middle course. The upper portion of the front tube sheet is stayed in the usual manner with diagonal braces.

Fire box brackets upon either side of the boiler are bolted to the furnace course. This often requires the local use of flush stay bolts. The front end of the boiler is supported by a pedestal riveted to the smoke box extension. Small handholes are installed over the fire door and at the corners of the water legs giving complete access to all sides of the fire box. A blow-off pipe is attached to the lower part of the throat sheet. The grates are supported by a bearer bolted to the mud-ring and sufficient room is provided below the tubes for a fire of ordinary thickness. The furnace must also have considerable height to secure space for the mingling and combustion of the gases over the fire.

79. Specifications. — The following general specifications are intended to apply to the design of any locomotive type boiler for contractors' use:

Shell Plates. — Plates shall be of best quality O. H. Firebox Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	52,000
Not more than	62,000
Yield point, min. lbs. per sq. in	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T.S.}}$

Heads. — Heads shall be of best O. H. Flange Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	55,000
Not more than	65,000
Yield point, min. lbs. per sq. in	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T.S.}}$

All heads shall be machine-flanged by the spinning process to an inner radius not less than two times the plate thickness, and shall be thoroughly annealed.

Rivets. — Rivets shall be of the best quality of soft O. H. steel according to the following specifications:

Tensile strength, lbs. per sq. in.:	
Not less than	45,000
Not more than	55,000

Elongation in 8 ins., min. per cent, but need not exceed 30 per cent	$\frac{1{,}500{,}000}{\text{T.S.}}$
Shearing strength, lbs. per sq. in	45,000
Crushing strength (bearing pressure), lbs. per sq. in. of projected area	96,000

Rolling. — The plates shall be rolled cold by gradual and regular increments to the exact radius required and the whole circumference rolled to a true circle. Butt-straps shall be rolled to the same radius as the shell in special forms made for that purpose.

Planing. — The edges of the plates shall be beveled to an angle of about 75 degrees on a plate planing machine. After the heads have been flanged and annealed, the calking edge shall be turned off on a milling machine to the same bevel as the shell plates.

Seams and Riveting. — All external longitudinal seams shall be of the butt-joint type with inside and outside cover plates. The circular or girth seams in the shell shall be of the single-riveted lap-joint type. The fire box and mud-ring seams shall be of the single-riveted lap-joint type. The seam in the throat sheet where the latter joins the cylindrical shell shall be of the double-riveted lap-joint type. Longitudinal seams shall come well up in the steam space and be arranged to break joints in the several courses.

Rivets shall be $\frac{1}{16}$ in. smaller in diameter than the holes they are to fill. All holes shall be drilled to size with all plates and cover plates in place, after the plates have been rolled. After drilling the holes the plates shall be taken apart and the burrs removed from the edges of the holes. Previous to the drilling the holes may be punched cold to a size $\frac{1}{4}$ in. less than their finished diameter. All riveting shall be done with a hydraulic riveting machine whereever practicable, pressure being kept on each rivet until it has taken its shrinkage to insure tight joints. Rivet heads, on all machine driven rivets, shall be of the type ordinarily used in pressure work. Where necessary the rivets may be pneumatically driven, the formed head taking the above shape, the other retaining its original shape.

Calking. — All seams shall be carefully calked with a pneumatic hammer, using a round-nosed calking tool.

Tubes. — Tubes shall be made of the best American lap-welded or seamless steel, arranged in straight rows with a circulation space at the center of the boiler if possible. Tube holes shall be punched not over 1 in. in diameter, drilled to size with a revolving cutter and the edges chamfered to a radius of $\frac{1}{16}$ in. Tube ends shall be carefully expanded and beaded over with pneumatic tools.

Water Legs. — Water legs at least 3 ins. in net width shall be provided at all sides of the fire box. A forged or cast steel mud-ring shall seal the bottom of the water legs and provide a surface against which the inner and outer sheets may be calked.

Fire Box. — A symmetrical rectangular fire box shall be provided, giving ample grate area and space for the combustion of the fuel at a reasonable rate. The surface of the grate shall come at least 2 ins. above the upper surface of the mud-ring. Sufficient room shall be provided between the surface of the grate and the lowest tubes to permit the use of a heavy fuel-bed without danger of clogging the tube holes. The furnace roof and side sheets shall be made without a seam, rolled from one piece of plate. The furnace roof shall be curved to a large radius or left flat as per previous agreement.

Staying in Front Tube Sheet. — The front head above the tubes shall be stayed with diagonal braces and the arrangement shall be such as to secure an even spacing of the stayed points and a good support for the entire surface. The reaches of plate between stay rivets shall not exceed in length those allowed for stay bolts on flat plates of similar thickness. The braces shall be so spaced as to secure the most uniform loading possible. The braces shall be of weldless steel and of approved design. The angle between the brace and shell shall in no case exceed 20 degrees and suitable anchorage shall be provided where the braces join the shell.

Staying in Water Legs. — Solid steel stay bolts with 12 threads per inch shall be used for staying the flat sheets at the sides of the furnace. The thread shall be continuous through both sheets and after screwing in place the stay bolt end shall be securely riveted over by pneumatic hammer. The spacing of such stay bolts shall conform to the requirements of the thinner sheet if the two sheets joined are of different thickness. The stay bolts shall be spaced in regular order and shall secure a good support for the entire surface.

Staying of Furnace Roof. — The staying of the crown sheet above the furnace shall be either by means of radial stays or slung crown bars as per previous agreement. For radial stays the spacing shall be regular throughout, all stays being placed radial to the furnace roof. When the angle at which the radial stay meets the shell is less than 75 degrees, a double thickness of plate shall be employed at the anchorage in the outer shell. No inherent strength shall be assumed from the curvature of the furnace roof, the stayed points being spaced the same as in the water legs. In the case of slung crown bars ample strength shall be provided in the links, pins and angles used to safely sustain the entire load upon the furnace roof.

Throat Staying. — The space occasioned by the removal of tubes to make room for the handhole in the front tube sheet shall be stayed by the insertion of one or more weldless steel boiler braces in the lower part of the rear tube sheet. The arrangement of this staying shall be such as to keep all reaches of unsupported plate within safe limits and to give a secure support for the entire area. To provide circulation such brace heads shall be set off from the tube sheet a distance not less than 3 ins. by the use of 1 in. pipe thimbles inserted upon the rivets.

Staying in Rear External Furnace Sheet. — The segment of plate above the water leg surrounding the fire door shall be supported either by weldless steel boiler braces anchored to the shell or by vertical angles in pairs slung from the boiler shell by eye bolts. In the case of diagonal braces the same specifications shall obtain as in the front tube sheet. The brace shanks shall run forward to the shell with a maximum deviation from a radial position of not more than 5 degrees. A secure anchorage shall be provided at the shell and care shall be exercised that brace rivets are not in close proximity to radial stay bolt holes. In the case of slung angles one or more eye bolts may be used to anchor the same to the shell. Such eye bolts shall have abundant area to support the share of the load from the stayed plate which naturally comes upon them, no assistance being assumed from the inherent stiffness of the flat sheet.

Manhole in Shell. — One manhole 11 ins. × 15 ins. shall be located in the shell on top of the boiler in the center of the middle course. It shall have a pressed steel frame or ring riveted to the shell. It shall be fitted with a pressed steel cover, bolt and yoke, and a copper or rubber gasket. The frame shall have a net cross-sectional area on a line through its center parallel to the axis of the shell, the tensile strength of which is at least equivalent to that of the shell plate removed on the same line. The frame shall be riveted to the shell with two rows of rivets. The resistance of the rivets in shear or in crushing on one side of the longitudinal section through the center of the frame shall not be less than the tensile resistance of the plate removed on the same section.

Handholes. — Five 3 ins. × 4½ ins. handholes shall be installed in the external furnace sheets of the boiler, four being placed at the bottom of the water legs and one over the fire door near the furnace roof. The handholes shall be so arranged as to give complete access to all the sheets forming the fire box of the boiler. There shall be at least 1 in. of flat plate in the clear around each handhole opening. Provision shall be made for the use of a standard cast steel handhole cover and yoke.

One handhole 4 ins. × 6 ins. in size shall be located in the lower part of the front tube sheet so placed as to give access to the interior of the shell below the tubes.

Nozzles. — Two cast or forged steel nozzles of diameter specified shall be machine-riveted to the shell at the top of the front course for main steam pipe and safety valve connections. The flanges which join the shell shall be beveled and calked on the outside.

Water Column Connections. — Two holes of diameter specified shall be drilled in the furnace course near the rear end and reinforced with forged steel pipe flanges to provide for the water column connections. The water connection shall be taken from a point in the right water leg of the shell about one foot below the center of the boiler, and located about 12 ins. from the rear external furnace sheet. A similar opening at the top of the shell shall be provided for the steam connection.

Feed Pipe Connections. — Upon the right side of the boiler at a level a little below that of the top row of tubes and at the center of the middle course a forged steel pipe flange shall be riveted to the shell. The size of the flange shall be sufficient to accommodate a suitable brass feed pipe bushing. The feed pipe shall be screwed into this bushing from without and a male and female elbow with distributing pipe from within. The latter shall extend about 4 ft. along the side of the boiler toward the front, its front end being capped and its lower half perforated to throw the feed water into the circulation.

Blow-off Connections. — A forged steel pipe flange or pad shall be riveted to the throat sheet just above the mud-ring and tapped to receive the size of blow-off pipe specified.

Mud-Ring. — A welded or cast steel mud-ring 3 ins. wide and at least 2½ ins. high shall be provided to seal the water legs of the boiler. The same shall be securely riveted to the inside and outside sheets with a pitch of rivets corresponding to the thickness of the thinner sheet. At the corners the mud-ring shall be forged both inside and out to fit the necessary overlap of the sheets. Boiler patch-bolts screwed and riveted to place may be employed upon the outer radius of the mud-ring corners.

Smoke Box. — Provision shall be made in an extension of the front course for a smoke box. The area of the uptake leading therefrom shall be approximately one-eighth that of the grate.

Fusible Plug. — A fusible plug of approved pattern filled with pure tin shall be located in the highest portion of the furnace roof near the rear tube sheet flange.

Supports. — Two cast steel brackets of approved design shall be attached to each side of the rear course so as to support the boiler upon skids. The brackets may be attached to the water legs by extra long stay bolts, the same being securely riveted after screwing to place. The steel brackets shall be thoroughly annealed and tightly calked around their edges to prevent leakage. In case this is undesirable a thin gasket of soft steel shall be fastened between the shell and the bracket and calked tight. The pedestal forming the front support shall be a steel casting designed to give adequate strength for connecting the skids at the front end and supporting the cylindrical shell of the boiler at the same time. The shell shall rest in the cradle thus formed. The brackets and pedestal shall be securely bolted to the skids upon which the boiler is to be transported.

Test. — A hydrostatic pressure test (cold water) of one and one-half times the working pressure for which the boiler is intended, shall be applied, and the boiler shall be suitably designed for tight joints under that pressure.

80. Statement of Problem. — The solution of the following problem will be given in full.

Design a locomotive type boiler for contractors' use with smoke box in the shell extension, to conform to the preceding specifications and to embody the following general dimensions:

General Dimensions:

Rated Horse-Power (A.S.M.E. standard)	125
Working Pressure, lbs. per sq. in.	150
Length of Tubes, ft.	15

Number of Courses	3
Diameter of Steam Nozzles (2), ins	4½
Diameter of Feed Pipe, ins	1¼
Diameter of Blow-off Pipe, ins	2
Diameter of Water Column Connections, ins	1¼
Kind of Coal	Bituminous
Types of Joints to be used:	
Ring Seam Joints	Double-Riveted Lap.
Firebox Joints	Single-Riveted Lap.
Longitudinal Joints	Butt-joint with inside and outside cover plates, having an efficiency of at least 94 per cent.

Design to include:

(*a*) Complete calculations.

(*b*) Working drawings fully dimensioned, of

(1) Tube sheets, water leg and fire box with staying. Scale 3 ins. =1 ft.

(2) Boiler: end view, longitudinal section and development of joints. Scale 1½ ins. = 1 ft.

The following specific values of constants, for which a general discussion i given in Chap. IV, will be assumed:

Rate of Evaporation, lbs. of water per lb. of coal, about	9
Rate of Combustion (bituminous coal), lbs. per sq. ft. of Grate Surface per hr., about	17.5
Ratio: Length to Outside Diameter of Tubes, about	60 to 1
Ratio: Internal Transverse Tube Area to Grate Area, about	1 to 6
Steam Space per Boiler H.P., cu. ft., about	0.80
Ratio: Steam Space to Water Space, about	1 to 2.5
Heating Surface per Boiler H.P., sq. ft., at least	10
Ratio: Heating Surface to Grate Surface, about	45 to 1
Mean Water Level above Crown Sheet, ins., at least	6
Boiler Diameters to vary by increments of, ins	6
Reach of Riveting Machine, ft	8
Grate Width to be consistent with vertical water legs, the latter having a net width of, ins	3
Least Factor of Safety	5

81. Calculations.

Grate. — The number of pounds of water to be evaporated per hour may be obtained from the rated horse-power (A.S.M.E. standard, p. 5).

$$125 \times 34.5 = 4312.5 \text{ lbs. of water per hour.}$$

The coal to be burned per hour is found from the above and the rate of evaporation given.

$$\frac{4312.5}{9} = 479.2 \text{ lbs. of coal per hour.}$$

From the rate of combustion the theoretical grate area is found to be

$$\frac{479.2}{17.5} = 27.38 \text{ sq. ft.}$$

Trial size of grate 5 ft. wide by 5.5 ft. long which gives 27.5 sq. ft.

Tubes. — The tube diameter for the given ratio of length to outside diameter is

$$\frac{15 \times 12}{60} = 3 \text{ ins.}$$

Tubes 3 ins. in diameter will be used.

From Table XXIX, p. 190:

Internal circumference = 8.740 ins.,
External transverse area = 7.069 sq. ins.,
Internal transverse area = 6.079 sq. ins.

The total internal transverse tube area is computed from the theoretical grate area and the ratio given

$$\frac{27.38 \times 144}{6} = 657.1 \text{ sq. ins.}$$

The number of tubes to give this area is

$$\frac{657.1}{6.079} = 108.1.$$

The attempt will be made to lay out 110 tubes on the tube sheet drawing with suitable spacing for this type of boiler.

Boiler Diameter. — On account of the irregularity of the steam and water space in the furnace course it will be assumed that three-fourths of the total boiler volume is contained in the cylindrical portion. If, therefore, it is desired to obtain a final steam volume of 0.8 cu. ft. per horse-power, the diameter of the shell will be based upon a trial volume three-fourths this amount or 0.6 cu. ft. per horse-power.

Then

$$\begin{aligned}
&\text{Steam space, } 125 \times 0.6 &&= 75.0 \text{ cu. ft.}\\
&\text{Water space, } 75.0 \times 2.5 &&= 187.5 \text{ cu. ft.}\\
&\text{Tube space, } \frac{110 \times 180 \times 7.069}{1728} &&= 81.02 \text{ cu. ft.}\\
&\qquad\qquad\qquad\text{Total} && \ \overline{343.52} \text{ cu. ft.}
\end{aligned}$$

For a length of 15 ft. the corresponding area of the head will be

$$\frac{343.52 \times 144}{15} = 3298 \text{ sq. ins.}$$

This area corresponds to a diameter of 64.80 ins. To bring the latter to a practical figure and make some allowance in the capacity of the boiler, the diameter will be made 66 ins.

Water Level. — In anticipation of the use of radial stays the roof of the furnace is given a curvature corresponding to an inside radius of 84 ins. In order to insure 6 ins. of water over the furnace roof, the latter must be so located as to give the stipulated amount with respect to the mean water level. The total steam volume in the cylindrical portion as calculated above is 75 cu. ft. With tubes 15 ft. long the area of the steam segment will be

$$\frac{75}{15} = 5 \text{ sq. ft.} \quad \text{or} \quad 720 \text{ sq. ins.}$$

Then

$$\frac{A}{D^2} = \frac{720}{(66)^2}$$
$$= 0.1653.$$

From the plot, Fig. 118, p. 213,

$$\frac{h}{D} = 0.263$$

or

$$h = 17.36 \text{ ins.}$$

Adding to this 6 ins., the distance from the crown sheet to the top of the boiler should be $23\frac{3}{8}$ ins.

Thickness of Cylindrical Shell. — The desired shell thickness is readily obtained with the given data from the plot, Fig. 11, p. 57, where the joint efficiency is taken 94 per cent.

$$\text{Thickness} = 0.479 \text{ in.}$$

Use a thickness of shell equal to $\frac{1}{2}$ in. The tentative thickness of tube sheet will be assumed $\frac{1}{2}$ in. since the necessities in regard to staying are not severe and there is considerable liability to overheating in the fire box.

Tube Sheet Lay-out. — Before proceeding further with the calculations, the arrangement of tubes must be determined from the tube sheet drawing.

In order to provide for a thickness of fuel-bed of at least 12 ins., the pitch line of the mud-ring will be assumed approximately 52 ins. below the center of the cylindrical shell. This will give plenty of room for the location of the grate level well above the surface of the mud-ring. With the necessary allowances for the circulation and spacing it is found that 110 tubes can be accommodated without difficulty in the tube sheet if placed upon diagonal rows. This number will give a slight increase in the heating surface which is very desirable in this type of boiler.

Final Ratios. — Having established the steam and water volumes of the boiler, the ratios called for upon the tube sheet drawing, Fig. 194, will next be determined.

The dimensions of the grate previously assumed, 5 ft. × 5.5 ft., giving an area of 27.5 sq. ft., will be adopted since they fit well the diameter of the boiler and the width of water leg chosen.

The heating surface contained in the tubes is as follows:

$$\frac{110 \times 8.74 \times 15}{12} = 1201 \text{ sq. ft.}$$

To find the heating surface inside the fire box the latter will be assumed to be 5 ft. wide, 5.5 ft. long and 4.5 ft. high. The internal area of the four sides and top will, therefore, be

Top, 5 × 5.5		= 27.5 sq. ft.
Two sides, 2 × 5.5 × 4.5		= 49.5 sq. ft.
Front and rear ends, 2 × 5 × 4.5		= 45.0 sq. ft.
	Total	122.0 sq. ft.

From this must be deducted about 5.5 sq. ft. for the area of the tube holes and 1.5 sq. ft. for the fire door opening, making a net heating surface in the fire box of 116 sq. ft. Then the total heating surface for the boiler will be

$$1201 + 115 = 1316 \text{ sq. ft.}$$

The heating surface per H.P. will then be

$$\frac{1316}{125} = 10.53 \text{ sq. ft.},$$

an amount well in excess of that required.

The ratio of the heating surface to the grate area is

$$\frac{1316}{27.5} = 47.9,$$

which is satisfactory.

The relation between the transverse internal area of the tubes and the grate surface is as follows:

$$\frac{110 \times 6.079}{27.5 \times 144} = \frac{1}{5.92},$$

which is close to that called for in the specifications.

The final volumes of steam and water must next be corrected for the irregularities due to the arrangement of the fire box. The total length of the boiler is as follows:

$$\text{Tubes} + \text{fire box} + \text{water leg} = \text{Total length.}$$
$$15\text{ ft.} + 5.5\text{ ft.} + 3\text{ ins.} = 20.75\text{ ft.}$$

The area of the steam segment was formerly found to be 5 sq. ft., hence the total volume for the above length of boiler is

$$20.75 \times 5 = 103.75\text{ cu. ft.}$$

and the steam volume per H.P. is

$$\frac{103.75}{125} = 0.83\text{ cu. ft.}$$

This figure is fully equal to the quantity usually employed in horizontal return tubular boilers, hence it will be considered satisfactory.

The final water volume is made up of that contained in the cylindrical shell plus the volume contained in the water legs and over the roof of the fire box.

The area of a circle 5.5 ft. in diameter is 23.76 sq. ft. From this must be deducted the area of the steam segment, 5 sq. ft., and the combined transverse external area of the tubes. The latter quantity is

$$\frac{110 \times 7.069}{144} = 5.40\text{ sq. ft.}$$

The net transverse area occupied by the water in the cylindrical shell is

$$23.76 - (5 + 5.40) = 13.36\text{ sq. ft.}$$

The total volume of water in the cylindrical shell is, therefore,

$$13.36 \times 15 = 200.4\text{ cu. ft.}$$

The transverse area between the mean water level and the line *ab*, Fig. 194, where the parallel sides of the water legs begin, is found by the use of the planimeter upon the drawing to be 525.4 sq. ins. The volume corresponding to this area for the length of the furnace course is

$$\frac{525.4 \times 5.75}{144} = 20.98\text{ cu. ft.}$$

The distance from the surface of the mud-ring to the line *ab*, where the curvature of the crown sheet commences, measures from the drawing approximately 51 ins. and the distance from the former point to the average level of the furnace roof is 57 ins. The total volume in the water leg containing the fire door opening is, therefore, approximately,

$$\frac{3 \times 66 \times 57}{1728} = 6.53 \text{ cu. ft.}$$

Deducting from this the volume occupied by the door opening, assumed 0.5 cu. ft., the net volume of the rear water leg is approximately 6 cu. ft. The volume of the water leg at the throat calculated similarly is 2.83 cu. ft. The volume of the two side water legs is

$$\frac{2 \times 3 \times 51 \times 66}{1728} = 11.6 \text{ cu. ft.}$$

The total water volume in the boiler is, therefore, as follows:

Shell	200.4 cu. ft.
Over furnace	20.98 cu. ft.
Rear water leg	6.0 cu. ft.
Throat leg	2.83 cu. ft.
Two side legs	11.61 cu. ft.
Total	241.82 cu. ft.

The ratio of the steam to water volume is, therefore,

$$\frac{S}{W} = \frac{103.75}{241.82}$$

$$= \frac{1}{2.33},$$

which is sufficiently close to the ratio given in the specifications.

Disengaging Surface per Horse-Power. — With the tubes located in correct relation to the mean water level, the value of

$$\frac{h}{R} = \frac{17.38}{33}$$

$$= 0.527.$$

From the plot of Fig. 119, p. 216, this ratio gives a value of

$$\frac{w}{R} = 1.766$$

or

$$w = 58.28 \text{ ins.}$$

The total disengaging surface for an inside boiler length of 249 ins. is

$$\frac{58.28 \times 249}{144} = 100.8 \text{ sq. ft.}$$

and the disengaging surface per H.P. is

$$\frac{100.8}{125} = 0.806 \text{ sq. ft.,}$$

a quantity well within the limits prescribed by Table XXXVIII, p. 215.

Staying. — Taking up the staying in the order given in the specifications the procedure is as follows:

Front Tube Sheet. — Assuming the thickness of the front tube sheet $\frac{1}{2}$ in., the maximum allowable plate spacing, Table XXXIX, p. 233, is 6.42 ins. Drawing the stiff line of the tubes and the flange curve at the margin of the sheet, the total height of segment to be stayed measures, from the drawing, 25.63 ins. Dividing this quantity by the plate spacing 6.42 indicates that four zones and three layers of diagonal braces will be necessary. Each zone will have an approximate height of $6\frac{3}{8}$ ins. Using rivets $\frac{7}{8}$ in. in diameter in the braces the area which may be supported by each rivet, at 150 lbs. per sq. in. pressure, is found in Table XL, p. 235, to be 24.05 sq. ins. The circumferential spacing of the rivets to give this area will, therefore, be

$$\frac{24.05}{6.39} = 3.77,$$

or about $3\frac{3}{4}$ ins. The arrangement shown in the drawing, Fig. 194, is obtained by trial and the several areas supported per rivet determined by use of the planimeter. It is evident that there is a fair distribution of load among rivets. The lower rivets come rather close to the tubes, but if placed much farther away there will be too much unstayed plate in that region.

Water Legs and Furnace Roof. — To avoid overheating, the crown- and side-sheets of the fire box will be made $\frac{7}{16}$ in. thick. The maximum plate spacing, from Table XXXIX, p. 233, for the given thickness and pressure is 5.62 ins. Starting at the pitch line of the mud-ring this distance is stepped up the water leg and around the crown-sheet to the mud-ring upon the other side of the furnace. Reducing this spacing to $5\frac{3}{16}$ ins. will make it an integral fit between these limits, the number of spaces from mud-ring to mud-ring being 33. The points thus indicated are the center lines of the stay bolts supporting the water legs and crown sheet. Lines drawn radial to the furnace roof will indicate the direction of the radial stays in that locality. It is evident by use of a protractor that the four radial stays on each side next above those in the water legs form angles with the shell tangents less than 75 degrees. Hence a double thickness of plate is used at their anchorages.

The diameter of stay bolt to be used may be determined from the area supported as follows:

Total load per stay bolt with the latter pitched on squares

$$5.62 \times 5.62 \times 150 = 4738 \text{ lbs.}$$

By reference to Table XLVI, p. 264, it is seen that stay bolts $1\frac{1}{8}$ ins. in diameter are necessary and that there is a reasonable margin between their strength and the load to be supported. The longitudinal distance between the furnace ring seams is found from the drawing to be $61\frac{3}{8}$ ins. Inserting 11 stays in this length, the spacing is 5.11 ins. which is less than the plate limit given above.

If the above longitudinal spacing were continued below the center line of the boiler a wide unstayed space would result at each end of the furnace course. To obviate this, $6\frac{1}{4}$ ins., a quantity less than the maximum pitch for $\frac{1}{2}$ in. plate, is set off from each ring seam to the first stay bolt of the furnace course. The remaining length $55\frac{1}{2}$ ins. is stayed by bolts 5.05 ins. apart. The throat sheet and the rear external furnace sheet will be assumed $\frac{1}{2}$ in. in thickness, respectively. The maximum spacing which may be employed in reference to plate of this thickness is 6.42, Table XXXIX, p. 233. The stay bolts necessary in the latter case must sustain a load of

$$6.42 \times 6.42 \times 150 = 6182 \text{ lbs.}$$

and their diameter must be $1\frac{1}{4}$ ins. if a good margin of strength is to be allowed.

Throat Stays. — Proceeding according to the principles explained in relation to Fig. 139, p. 248, it is evident that two Scully braces will be necessary at the throat. The arrangement shown in the drawing, Fig. 194, is found by trial. The areas indicated are determined by planimeter and are safe ones for thimbled rivets $\frac{15}{16}$ in. in diameter. The braces are made 24 ins. long and are anchored to the shell just in front of the throat. A diameter of shank of $1\frac{1}{8}$ ins. is used.

Rear External Furnace Sheet. — Starting at the pitch line of the mud-ring the stay bolts supporting the water leg around the fire door are spaced vertically with a pitch of $6\frac{7}{16}$ ins., a figure substantially equal to the maximum spacing derived from Table XXXIX. In order to make the horizontal spacing of these stay bolts symmetrical with regard to the vertical center line of the boiler a pitch of 6 ins. is used. The horizontal row near the furnace roof is arranged by trial to conform to the curvature of the crown sheet. The load per stay bolt with this arrangement will be

$$6\tfrac{7}{16} \times 6 \times 150 = 5790 \text{ lbs.},$$

a quantity somewhat lower than the allowable amount for $1\frac{1}{4}$ in. steel stay bolts, Table XLVI, p. 264. A fire door, Fig. 168, p. 285, with an internal opening 20 ins. wide by 15 ins. high is placed with its center $19\frac{1}{4}$ ins. above the pitch line of the mud-ring. This arrangement gives a symmetrical location with regard to the stay bolts and permits the fuel-bed to be of sufficient thickness as well. To conform to the elliptical outline of the fire door opening the stay bolts in close proximity thereto are rearranged as shown in the drawing, Fig. 194.

Angles. — The staying in the upper portion of the rear sheet is best accomplished by riveting heavy vertical angles to the plate and connecting them by forged diagonal eye bolts to the shell above the furnace. Selecting for trial two angles, 5 ins. × 4 ins. × $\frac{5}{8}$ in., and placing them back to back with a distance of $1\frac{7}{8}$ ins. in the clear, it is evident that the vertical stay bolt pitch lines may be prolonged to the top of the shell and that the rivets may be seated suitably with such an arrangement. Stay rivets 1 in. in diameter will be selected for trial. The maximum plate spacing in $\frac{1}{2}$ in. plate at 150 lbs. per sq. in. pressure is 6.42 ins., Table XXXIX, p. 233. The maximum allowable area per rivet with the diameter and pressure at hand is 31.42 sq. ins., Table XL, p. 235. An approximate vertical spacing may be found by dividing the load area by the plate spacing, or

$$\frac{31.42}{6.42} = 4.895 \text{ ins.}$$

From the latter figure a vertical spacing varying from $4\frac{3}{4}$ ins. to $4\frac{3}{8}$ ins., as shown on the drawing, Fig. 194, is assumed. The plate supported by the several stay rivets is determined by the method of loci given in Fig. 139, p. 248, and by use of a planimeter the corresponding areas and loads are found. The angles are prolonged toward the top of the shell to the limit of flat plate, in order that as staunch an arrangement as possible may be secured. At the same time the two diagonal eye bolts will be designed of suitable size to sustain the sum of all the rivet loads in order to prevent overloading of the flange by the angles.

Rod Shanks. — The areas assignable to the several rivets are set down upon the drawing, Fig. 194. Multiplying each of these by the working pressure, the total load upon the pair of angles near the vertical center line of the boiler is found to be 35,000 lbs., and that upon the angles remote from the center line 25,200 lbs. To avoid mistakes in assembling the boiler parts all the diagonal eye bolts will be made of the same diameter. Inasmuch as the diagonal eye bolts may run at an angle of 20 degrees with the shell the gross load per bolt under the worst conditions will be

$$\frac{35{,}000}{2 \times \cos 20^\circ} = 18{,}620 \text{ lbs.}$$

By reference to Table XLIV, p. 246, it is evident that a rod shank $1\frac{3}{4}$ ins. in diameter will be required. This size will be used for anchoring all the angles to the shell. At the left side the stay rivets do not properly support all of the plate, hence a Scully brace with shank diameter $1\frac{1}{8}$ ins. and head containing two $\frac{7}{8}$ in. rivets, is inserted as shown. The load upon these rivets is within safe limits. The diagonal eye bolts reach forward at angles of 15 and 20 degrees, respectively, with the shell.

Pins. — The pin by which the eye bolt is attached to the angles is calculated as a beam supported at the ends and having a distributed load equal to the gross rod pull, 18,620 lbs. Substituting in the usual beam formula

$$M = f\frac{I}{y},$$

and using a modulus of rupture about 1.8 times the tensile strength, the value of the pin radius may be found. The span of the theoretical beam is assumed to reach to the centers of the angle legs which form the supports.

Then, using a factor of safety of 6,

$$\frac{18{,}620 \times 2.5}{8} = \frac{60{,}000 \times 1.8 \times \pi r^3}{6 \times 4}.$$

Solving, $r = 0.744$ in. and the pin diameters should be taken $1\frac{1}{2}$ ins.

Rod Eyes. — Following the procedure set down in regard to link eyes, p. 256, the area through the pin center will be made to exceed that of the rod shank by 50 per cent. Assuming the width of eye as $1\frac{7}{8}$ ins. to correspond to the space between angles, it is easily found that the diameter of eye must be $3\frac{1}{2}$ ins.

Pads. — The pad at the shell end is assumed of the proportions shown and is made to seat six $\frac{7}{8}$ in. rivets. The shearing load per sq. in. on these rivets will be

$$\frac{17{,}500}{6 \times 0.601} = 4850 \text{ lbs.},$$

which represents a factor of safety of about 9 and is therefore a safe stress. The bearing pressure upon the rivets, for a shell thickness of $\frac{1}{2}$ in., is

$$\frac{17{,}500}{6 \times 0.875 \times 0.5} = 6670 \text{ lbs. per sq. in. of projected area.}$$

The latter figure represents a factor of safety of about 15 and consequently will not cause local distortions in the shell. It is to be noted in the arrangement of rivets in the angles that a margin of about 6 ins. is preserved between the outer rivets and the shell. This is necessary in order to permit some flexibility. The assumption that the diagonal eye bolts carry all the load from the angles is doubtless very far upon the safe side. In some cases it may be wise to choose eye bolts somewhat less than the diameter given in the table in anticipation of the fact that a good portion of the load will be transmitted directly to the shell. The assumption given above, however, is believed to be a correct one for cases of this kind in general.

Riveted Joints. — There are four riveted joints demanding attention in this boiler, *viz.*: the longitudinal seams, the shell ring seams, the throat joint and the fire box seams. The latter only are exposed to the fire and hence their calking limits must be established by reference to the values given upon p. 132.

In selecting the longitudinal joint it is to be noted that an efficiency of at least 94 per cent is required. Therefore, joint *O*, the proportions of which are given in Table XXI, p. 171, will be selected. With the thickness of shell $\frac{1}{2}$ in. this joint gives a range of efficiencies slightly higher than that called for.

The following data is necessary in ascertaining the pitch and laps of the various joints:

Thickness of shell	$\frac{1}{2}$ in.
Thickness of tube sheets	$\frac{1}{2}$ in.
Thickness of fire box, sides and top	$\frac{7}{16}$ in.
Thickness of throat sheet	$\frac{1}{2}$ in.
Thickness of fire door sheet, external	$\frac{1}{2}$ in.
Thickness of inside cover plate	$\frac{7}{16}$ in.
Thickness of outside cover plate to be governed by the necessities in regard to calking.	

With a rivet $\frac{15}{16}$ in. in diameter the pitch, as taken from Table XXI, p. 171, is 17.42 ins. and the corresponding efficiency 94.6 per cent.

The calking pitch along the outer row will, therefore, be

$$\frac{17.42}{4} = 4.355 \text{ ins.}$$

Referring to Table XXIII, p. 173, it is noted that the calking limit for a working pressure of 150 lbs. per sq. in. with $\frac{15}{16}$ in. rivets is 4.57 ins. and that with this figure an outer cover plate at least $\frac{1}{2}$ in. in thickness must be used. The joint will, therefore, be made up with such an outside cover plate since the thickness of the latter does not affect the pitch. The rivets in the outer rows of this joint fail by shearing and those upon the inner by crushing, hence the following lap values may be determined.

Inside cover plate.	Table III.	Lap $=1.42$ ins.	Use $1\frac{7}{16}$ ins.
Outside cover plate.	Fig. 76, $\frac{t}{t_1} = 1$.	Lap $= 1.23$ ins.	Use $1\frac{1}{4}$ ins.
Main plate.	Table IV.	Lap $= 1.54$ ins.	Use $1\frac{9}{16}$ ins.

The distance between the staggered rows along the inside of this joint is $2\frac{1}{4}$ ins., found graphically by the method of Fig. 52, p. 101. The next outer row of rivets will be placed $1\frac{1}{4}$ ins. from the lap edge to provide room for driving the rivets, and the outermost row $2\frac{1}{4}$ ins. farther from the joint center.

The ring seams in the shell will be single-riveted lap joints. The pitch of $\frac{15}{16}$ in. rivets in $\frac{1}{2}$ in. plate, Table VII, p. 157, is 2.07

ins. and the corresponding efficiency 54.6 per cent. This is ample for the requirements at hand.

On account of expansion and contraction from its proximity to the fire box, the throat seam is under severe stresses in locomotive type boilers. Therefore this joint, embracing the lower semi-circumference of the shell, will be double-riveted and its ends calked into the ring seam where the furnace course joins the middle course. The pitch corresponding to the conditions at hand, Table VIII, p. 158, is 3.20 ins. and the efficiency 70.7 per cent, both of which are adequate for this seam. The distance between diagonal rows, as determined graphically from the method given in Fig. 52, p. 101, is 1.85 ins. Use $1\frac{7}{8}$ ins. The value of the lap from Table III, p. 137, is 1.36 ins. Use $1\frac{3}{8}$ ins.

The pitch of the hot seams joining the furnace sheets will be based upon a thickness of plate of $\frac{7}{16}$ in. In order to present less metal to the destructive action of the fire $\frac{7}{8}$ in. rivets will be used in all the fire seams. Referring to p. 132, it is noted that the distance between rivets, $p - d$, under the given conditions may be $1\frac{3}{8}$ ins. Hence the pitch to be used will be

$$\begin{aligned} p &= \tfrac{7}{8} + 1\tfrac{3}{8} \\ &= 2\tfrac{1}{4} \text{ ins.} \end{aligned}$$

The theoretical pitch under similar conditions, Table VII, p. 157, is 2.00 ins. Consequently the laps to be used will be based upon the shearing of the rivets rather than upon the tearing of the plate. Referring to Table III, p. 137, the following lap values are noted:

For $\frac{7}{16}$ in. plate. Lap = 1.30 ins. Use $1\frac{5}{16}$ ins.
For $\frac{1}{2}$ in. plate. Lap = 1.24 ins. Use $1\frac{1}{4}$ ins.

All the seams in the fire box, including that at the mud-ring, will use the pitch just calculated since such a value is safe against leakage and presents a minimum amount of metal to the action of the fire.

Insertion of Seams. — The length of plate between the ring seams at the throat and smoke box, as determined from the drawing, Fig. 195, is 180.5 ins. With the thicknesses of plate at hand it will be possible to end the joints in the ordinary manner by tucking the outside cover plate under the succeeding course. Referring to joint O, Fig. 97, the inside pitch p_1, previously determined, was 4.355 ins. Then the end pitches p_2 will be

$$4.355 \times 0.8 = 3.484. \qquad \text{Use } 3\tfrac{3}{8} \text{ ins.}$$

There are four of these end pitches in the two joints to be installed, hence

$$3\tfrac{3}{8} \times 4 = 13\tfrac{1}{2} \text{ ins.}$$

will be deducted from the above length between seams, leaving a net distance for regular pitches equal to 167.0 ins. The number of small pitches comprised in this seam will be

$$\frac{167.0}{4.355} = 38.34.$$

Assuming that 38 pitches will be used, the actual short pitch will be

$$\frac{167.0}{38} = 4.394 \text{ ins.}$$

This slight increase will not serve to alter the caloulations made in deriving the pitch. The corresponding long pitch p will be 17.58 ins. and the efficiency practically that of the original pitch, 94.6 per cent.

To avoid conflict with the staying the steam pipe and safety valve nozzles must be attached to the front course, therefore the latter will be made somewhat wider than the middle one. If 21 small pitches are inserted in the front course and 17 in the middle, their respective pitch line lengths will be

Front course $21 \times 4.394 + (2 \times 3\tfrac{3}{8}) = 99.02$ ins. Use 99 ins.
Middle course $17 \times 4.394 + (2 \times 3\tfrac{3}{8}) = 81.45$ ins. Use $81\tfrac{1}{2}$ ins.

The odd pitch is inserted in each joint in order to make the relation between the outer and inner rows of rivets that shown in the drawing, Fig. 195. The objection is sometimes raised that the rivets in the outer row should not come immediately adjacent to those in the staggered rows. While it is doubted whether this distinction amounts to much, the joint is inserted with this criticism in view. Forty pitches could have been used, making two duplicate joints or twenty pitches each, the insertion being precisely that shown in Fig. 97.

The circumference of a circle 66 ins. in diameter is 207.3 ins. Dividing this by the ring seam pitch 2.07 ins., 100 spaces are indicated in the girth joints. The semi-circumference forming the throat seam will require

$$\frac{207.3}{2 \times 3.20} = 32.44 \text{ spaces.}$$

Reducing this to 32, the actual pitch to be used will be

$$\frac{207.3}{2 \times 32} = 3.244 \text{ ins.}$$

Mountings. — The drawing, Fig. 195, shows the attachment of the steam nozzles, manhole, feed and blow-off pipes, and water column connections called for in the specifications.

Uptake. — The uptake generally ranges between $\frac{5}{8}$ and $\frac{3}{4}$ of the boiler diameter in width, and comprises about $\frac{1}{8}$ the grate area. The necessary transverse area, therefore, for this case will be

$$\frac{27.5}{8} = 3.44 \text{ sq. ft.}$$

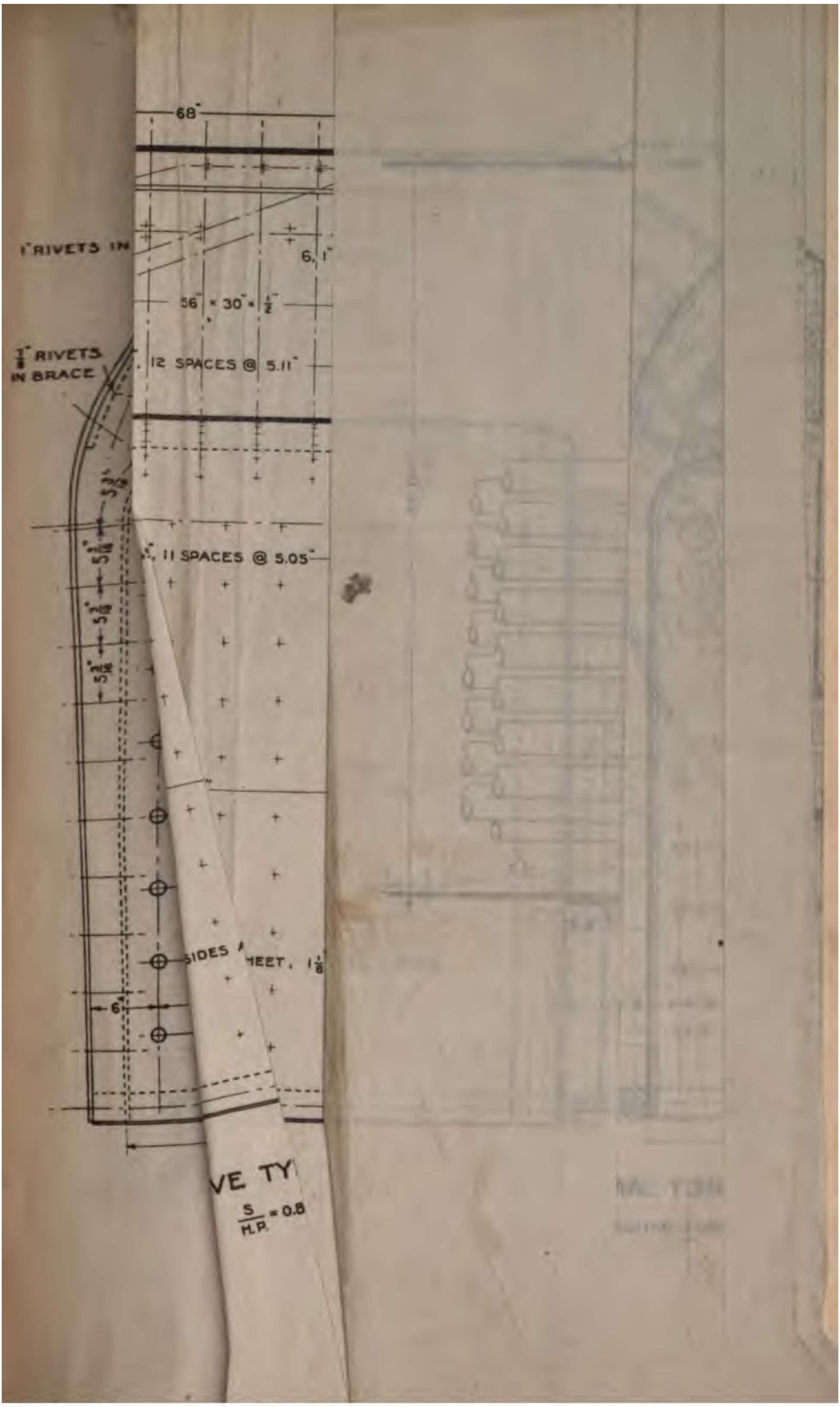
Assuming the width as 45 ins. the requisite depth will be

$$\frac{3.44 \times 144}{45} = 11 \text{ ins.}$$

With semi-circular ends, an outlet 12 ins. × 45 ins. will be used, and at least 2 ins. of clear plate will be secured around the opening to provide for the attachment of the smoke pipe. The cast iron smoke box door-frame is bolted to the smoke box extension by small eye bolts.

Fusible Plug. — The location of the fusible plug in the crown sheet near the front end of the fire box is required by law. If there are stay bolt holes along the center of the furnace roof the fusible plug may be inserted in the clear plate at the right or left.

Blow-off. — A pad of $\frac{1}{2}$ in. plate riveted to the lower portion of the throat sheet provides the specified blow-off connection.


68"
1" RIVETS IN
6, 1"
56" × 30" × ½"
⅞" RIVETS IN BRACE
12 SPACES @ 5.11"
3¾"
5 3/16"
5 3/16"
5 3/16"
11 SPACES @ 5.05"
SIDES
HEET, 1⅛"
6"
VE TY
S/H.P. = 0.8

CHAPTER IX.

TANK DESIGN.

TANKS may be divided into two general classes, *viz.*, rectangular tanks and cylindrical tanks. The former, being in reality a series of flat plates, are necessarily weak and difficult to make tight. Even if of moderate size they require thorough bracing. This may be accomplished by through stay rods in conjunction with structural shapes securely riveted to the sides, or, where through rods cannot be used on account of the solidity of the contents of the tank, such as in the case of tallow tanks, the sides may be stiffened by riveting structural shapes directly to the shell plates.

Such tanks are rarely used where the pressure within is other than that due to the weight of liquid contained. Rectangular pressed steel tanks made in sections with pressed ribs are on the market* and combine strength with lightness. The sections are four feet square and are bolted together, the joints being made tight with lead strips $\frac{1}{8}$ in. thick placed between the flanges and calked from the inside.

Tanks to withstand high pressures are ordinarily made cylindrical in shape. Such tanks include stand-pipes, water-towers, vulcanizers, dyeing vats, bleaching kiers, digesters, cold and hot water tanks, etc. The principles laid down for the design of steam boilers hold equally well in these cases so far as strength, riveted joints and staying are concerned. The complications due to contact with the flames are not met, however.

Cylindrical tanks as liquid containers are usually placed with their axes either vertical or horizontal. When vertical the capacity is readily calculated for any given depth. When the axis is horizontal the capacity is not as easily determined, and Table LIV has been prepared to assist in obtaining the contents of such tanks. The table reads in cubic feet per foot of length for every inch of depth, the tanks varying from 18 ins. to 96 ins. in diameter

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* American Spiral Pipe Works.

TABLE LIV.

CAPACITY OF HORIZONTAL CYLINDRICAL TANKS.

In Cubic Feet per Foot of Length.

Diameter of Tank — Inches.

Depth In inches.	18	24	30	36	42	48	54	60	66	72	78	84	90	96
Full	1.767	3.142	4.909	7.069	9.621	12.57	15.90	19.64	23.76	28.27	33.18	38.49	44.18	50.27
1	0.039	0.045	0.050	0.055	0.060	0.064	0.067	0.071	0.075	0.079	0.082	0.083	0.087	0.090
2	0.107	0.125	0.140	0.154	0.167	0.179	0.190	0.200	0.215	0.220	0.227	0.240	0.246	0.256
3	0.193	0.227	0.256	0.288	0.305	0.326	0.346	0.368	0.387	0.403	0.423	0.436	0.450	0.467
4	0.292	0.344	0.389	0.429	0.466	0.499	0.533	0.560	0.590	0.616	0.646	0.666	0.692	0.717
5	0.400	0.474	0.537	0.595	0.646	0.693	0.739	0.783	0.823	0.853	0.896	0.936	0.967	0.998
6	0.516	0.614	0.699	0.773	0.844	0.907	0.966	1.023	1.074	1.126	1.174	1.220	1.265	1.306
7	0.636	0.762	0.871	0.967	1.054	1.134	1.209	1.280	1.349	1.411	1.474	1.534	1.580	1.645
8	0.759	0.917	1.051	1.170	1.278	1.376	1.470	1.555	1.636	1.717	1.791	1.862	1.935	1.997
9	0.884	1.076	1.239	1.382	1.512	1.630	1.741	1.848	1.944	2.041	2.129	2.215	2.300	2.375
10	.....	1.239	1.432	1.602	1.756	1.897	2.029	2.148	2.265	2.379	2.484	2.582	2.682	2.771
11	.....	1.404	1.630	1.829	2.008	2.171	2.324	2.468	2.601	2.732	2.852	2.974	3.082	3.194
12	.....	1.571	1.834	2.063	2.268	2.457	2.630	2.795	2.949	3.092	3.236	3.376	3.498	3.629
13	.....	.....	2.038	2.300	2.536	2.748	2.946	3.133	3.308	3.477	3.633	3.788	3.931	4.071
14	.....	.....	2.246	2.543	2.808	3.049	3.272	3.483	3.677	3.865	4.043	4.214	4.380	4.538
15	.....	.....	2.454	2.788	3.085	3.354	3.604	3.841	4.055	4.269	4.466	4.660	4.836	5.018
16	.....	.....	.....	3.035	3.367	3.666	3.945	4.203	4.445	4.678	4.897	5.111	5.309	5.504
17	.....	.....	.....	3.284	3.651	3.983	4.290	4.576	4.842	5.097	5.341	5.573	5.792	6.003
18	.....	.....	.....	3.534	3.939	4.303	4.639	4.956	5.247	5.528	5.792	6.046	6.286	6.522
19	.....	.....	.....	.....	4.230	4.628	4.993	5.341	5.661	5.964	6.253	6.531	6.794	7.053
20	.....	.....	.....	.....	4.520	4.956	5.355	5.731	6.079	6.406	6.722	7.023	7.305	7.591
21	.....	.....	.....	.....	4.811	5.285	5.720	6.126	6.501	6.860	7.200	7.526	7.827	8.136
22	.....	.....	.....	.....	.....	5.615	6.087	6.524	6.931	7.317	7.684	8.032	8.362	8.686
23	.....	.....	.....	.....	.....	5.948	6.458	6.928	7.367	7.782	8.175	8.551	8.902	9.256
24	.....	.....	.....	.....	.....	6.283	6.830	7.337	7.802	8.249	8.670	9.070	9.454	9.831

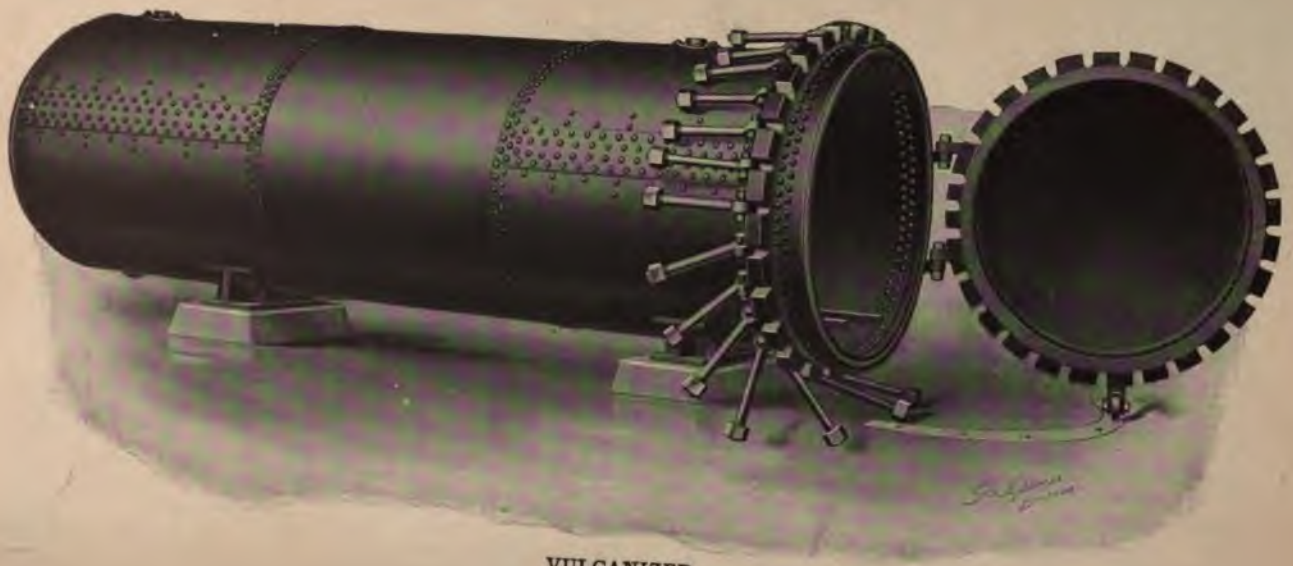
TABLE LIV.—(*Continued.*)

CAPACITY OF HORIZONTAL CYLINDRICAL TANKS.

In Cubic Feet per Foot of Length.

Diameter of Tank—Inches.

Depth in inches.	18	24	30	36	42	48	54	60	66	72	78	84	90	96
Full	1.767	3.142	4.909	7.069	9.621	12.57	15.90	19.64	23.76	28.27	33.18	38.49	44.18	50.27
25	.....	.....	.....	.....	.....	.....	7.203	7.743	8.247	8.722	9.173	9.598	10.00	10.41
26	.....	.....	.....	.....	.....	.....	7.577	8.156	8.694	9.200	9.678	10.14	10.57	11.00
27	.....	.....	.....	.....	.....	.....	7.952	8.570	9.141	9.683	10.20	10.69	11.14	11.59
28	.....	.....	.....	.....	.....	.....	.....	8.987	9.592	10.17	10.72	11.23	11.72	12.20
29	.....	.....	.....	.....	.....	.....	.....	9.402	10.05	10.66	11.23	11.78	12.30	12.81
30	.....	.....	.....	.....	.....	.....	.....	9.818	10.50	11.15	11.75	12.34	12.89	13.42
31	.....	.....	.....	.....	.....	.....	.....	.....	10.96	11.64	12.29	12.90	13.48	14.04
32	.....	.....	.....	.....	.....	.....	.....	.....	11.42	12.14	12.81	13.46	14.08	14.67
33	.....	.....	.....	.....	.....	.....	.....	.....	11.88	12.64	13.35	14.03	14.67	15.30
34	.....	.....	.....	.....	.....	.....	.....	.....	......	13.13	13.89	14.60	15.28	15.94
35	.....	.....	.....	.....	.....	.....	.....	.....	......	13.63	14.43	15.17	15.89	16.58
36	.....	.....	.....	.....	.....	.....	.....	.....	......	14.14	14.97	15.75	16.50	17.22
37	.....	.....	.....	.....	.....	.....	.....	.....	......	......	15.51	16.33	17.11	17.87
38	.....	.....	.....	.....	.....	.....	.....	.....	......	......	16.05	16.92	17.73	18.52
39	.....	.....	.....	.....	.....	.....	.....	.....	......	......	16.59	17.49	18.34	19.17
40	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	18.08	18.96	19.83
41	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	18.66	19.59	20.49
42	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	19.24	20.21	21.14
43	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	......	20.83	21.81
44	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	......	21.45	22.47
45	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	......	22.09	23.14
46	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	......	......	23.80
47	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	......	......	24.47
48	.....	.....	.....	.....	.....	.....	.....	.....	......	......	......	......	......	25.13



VULCANIZER.
The Bigelow Co.
FIG. 196.

by increments of 6 ins. When the tank is more than half full the column headed "Depth in ins." may be read "Distance of liquid level from top" and the contents determined by subtracting this figure from the capacity when full, as given in the first line.

When calculating the stress at the base of a vertical cylindrical tank care must be taken to note that the hoop stress is tension due to liquid pressure and the end-wise stress is compression due to the weight of the structure above. The true stress should then be obtained from the method outlined on page 58 and the design carried out accordingly.

The use for which a tank is intended will often determine the material of which it is to be made or with which it is to be lined. Tanks in which corrosion is objectionable are usually made of copper. To prevent chemical action between the steel plates and the liquid contained, it is often necessary to use linings of some such material as lead or silver. The method of securing these linings to the shell is explained in Chap. I, Art. 14. The design of the tank, however, is not affected by the fact that a lining is to be used.

When a cylindrical shell is to be driven at a high speed of rotation, as in the case of centrifugal dryers, it is often advisable to use similar joints on opposite sides to preserve a running balance.

To illustrate the general procedure in the design of pressure tanks, the complete calculations and drawings will be given for a vulcanizer.

82. Vulcanizer; *Description.* — The vulcanizer shown in Fig. 196 is typical of this class of pressure vessels. A horizontal cylinder consisting of three courses has longitudinal joints of the multi-riveted butt-strapped type and lapped joints for the girth seams. Since it is desirable that the inside of the vessel be free from staying the heads are of the dished or spherical form. The head on the closed end is riveted to the shell. The removable head is riveted to a flanged cast steel ring, forming a door which is hinged to a similar ring riveted to the shell. The flanges of both rings are slotted to receive locking bolts, the inner ends of which are permanently hinged to the ring on the shell and the outer ends provided with nuts. A packing ring is inserted on the face of the inner ring. A carriage and track relieve the hinge bolts of the weight of the head when open and facilitate operation. Two forged steel pipe flanges on top provide an inlet and outlet while a similar flange in the bottom at the rear is for use in draining the vulcanizer.

83. Statement of Problem.

Design a steel vulcanizer to conform to the following general dimensions and specifications:

General Dimensions:

Length, approximate, ft	24
Diameter, ins	60
Working Pressure, lbs. per sq. in	125
Number of Courses	3
Types of Joints:	
Ring Seams	Single-Riveted Lap.
Longitudinal Joint	Triple-riveted butt-joint with inside and outside cover plates, having an efficiency of at least 89 per cent.

Design to include:

(*a*) Complete calculations.

(*b*) Working drawings, fully dimensioned, showing a longitudinal section, development of joints and details of rings. Scale 1½ ins. = 1 ft.

Specifications:

Both heads shall be of the spherical form with a radius equal to the diameter of the shell.

The closed end shall be riveted directly to the shell. The removable head shall be riveted to a flange ring. A similar flange ring shall be riveted to the shell, and provision made for securely bolting the flange rings together.

The cover bolts shall be permanently hinged to the ring on the shell and shall be pocketed in slots on the rings.

The rings shall be provided with suitable lugs for hinge pins.

Forged steel pipe flanges shall be provided as follows:

One 6 inch flange at the center of each end course on top.

One 2 inch flange on the bottom at the center of the middle course and one 20 ins. from the back ring seam.

Materials.

Shell Plates:

The shell plates are to be of the best quality O. H. Firebox Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	52,000
Not more than	62,000
Yield point, min. lbs. per sq. in	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T.S.}}$

Heads:

The heads are to be of the best O. H. Flange Steel, having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	55,000
Not more than	65,000
Yield point, min. lbs. per sq. in	0.5 T.S.
Elongation in 8 ins., min. per cent	$\frac{1,500,000}{\text{T.S.}}$

Rings:

The rings shall be steel castings having the following qualities:

Tensile strength, lbs. per sq. in.:	
Not less than	60,000
Yield point, min. lbs. per sq. in	27,000
Elongation in 2 ins., per cent	22
Reduction of area, per cent	30

The castings are to be thoroughly annealed.

84. Calculations.

Thickness of Shell. — Assuming a tensile strength of 55,000 lbs. per sq. in. and a factor of safety of 5, Fig. 11, p. 57, may be used to obtain the shell plate thickness. For a diameter of 60 ins. and a joint of 89 per cent efficiency the thickness is found to be 0.385 in. To allow for corrosion the thickness will be increased to $\frac{7}{16}$ in.

Thickness of Heads. — The plate thickness for the dished heads obtained by the use of formula (2), p. 52, for thin spheres is 0.341 in. As shown in Chap. II, Art. 20, this thickness involves a small factor of safety and short life. The thickness used will be that obtained from Fig. 15, p. 67. The thickness is found to be 0.625 in. for the diameter and pressure given. Use $\frac{5}{8}$ in.

$f = \frac{pr}{2t}$

Bolts. — The size and number of bolts to be used in locking the removable head may next be calculated. Assume the diameter of the gasket to be 64 ins. Then under a test pressure of 187.5 lbs. or once and a half the working pressure, the total load on the bolts will be

$$187.5 \times 3217 = 603{,}000 \text{ lbs.}$$

If now an axial stress in the bolts of eighty times the wrench pull is assumed, as is frequently done in designs of this character, the number of bolts to resist the above load without leakage is readily determined. Since the test pressure causes the above load and an extra heavy pull on the wrench would be justified, the latter may be taken 250 lbs. The pull per bolt is then 20,000 lbs., and the number of bolts

$$\frac{603{,}000}{20{,}000} = 30 \text{ very nearly.}$$

As the gasket is of a more or less elastic material, the bolt load may exceed that due to tightening up, and a factor of safety of 10 is therefore to be used. This factor with an allowable tensile strength of 60,000 lbs. per sq. in. gives a working strength of 6000 lbs. per sq. in. With this conservative figure the size of bolts is calculated from the working pressure.

Hence $125 \times 3217 = 402{,}000$ lbs.

is the total load on 30 bolts and the root area of one bolt is

$$\frac{402{,}000}{30 \times 6000} = 2.23 \text{ sq. ins.}$$

The nearest practical size is a 2 in. bolt having a root area of 2.30 sq. ins.

To check the necessary wrench pull on a 2 in. bolt where Q = wrench pull,

$$80 \times Q = 6000 \times 2.30,$$
$$Q = 172.5 \text{ lbs.}$$

Pin for Bolt. — The pin in the eye of the bolt may be taken as a beam uniformly loaded with the bolt pull necessary to hold the working pressure. The distance between supports is taken 3 ins., thus allowing ½ in. either side of the eye for the supports.

Then
$$f = \frac{My}{I},$$

where
$$M = \frac{WL}{8}$$
$$= \frac{402{,}000 \times 3}{30 \times 8}$$
$$= 5025 \text{ in. lbs.}$$
$$\frac{y}{I} = \frac{4}{\pi r^3}.$$

Assuming a modulus of rupture for round steel of 100,000 lbs. per sq. in. and a factor of safety of 6, the radius may be determined.

$$\frac{100{,}000}{6} = \frac{5025 \times 4}{\pi r^3}$$
$$= 0.724 \text{ in.}$$

Use pin 1⅝ ins. in diameter.

The thickness of metal forming the eye of the bolt is computed on the basis of 50 per cent more area on a cross-section through the eye than the root area of the bolt. Taking the width of the same as the diameter of the bolt, *viz.*, 2 ins., the thickness is found to be

$$\frac{2.30 \times 1.5}{2 \times 2} = 0.86 \text{ in.}$$

Make cross-section 2 ins. × $\frac{7}{8}$ in., which gives an outside eye diameter of $3\frac{3}{8}$ ins.

Rings. — The only calculation on the ring is that to determine the width of metal for the lugs. Dismissing the brackets, which, however, are a source of considerable strength, the portion between bolts may be figured as a cantilever. The width of the cantilever to be tested, allowing $\frac{1}{4}$ in. clearance in the bolt slots, is very nearly

$$\frac{\text{Circumference of a 70 in. circle (assumed)}}{30} - 2.25$$

or

$$\frac{219.91}{30} - 2.25$$
$$= 5.08 \text{ ins.}$$

Setting the bolts high enough to clear the rivet heads of the ring seam (say $1\frac{3}{4}$ ins.), the moment arm of the bolt pull on the section at the base of the slot will be $1\frac{3}{4}$ ins. The moment is then

$$\left(\frac{402{,}000}{30}\right) 1.75 = 23{,}450 \text{ in. lbs.}$$

The modulus of rupture may be taken as the tensile strength and a factor of safety of 5 is sufficient.

Hence

$$f = \frac{My}{I},$$

$$\frac{60{,}000}{5} = \frac{23{,}450 \times 6}{5.08\, h^2},$$

$$h = 1.52 \text{ ins.}$$

This dimension will be made $1\frac{3}{4}$ ins.

Riveted Joints. — For the longitudinal joint a butt-joint is to be selected which shall have an efficiency of at least 89 per cent with the thickness of shell previously determined, *viz.*, $\frac{7}{16}$ in. Consulting the tables of maximum pitches it is found that a triple-riveted butt-joint, with rivets arranged as in joint *M*, Table XIX,

p. 169, fulfils the requirements. With a rivet $\frac{7}{8}$ in. in diameter the maximum pitch is 8.11 ins. and the efficiency is 89.2 per cent.

The calking pitch is, then, one-half of this, or 4.06 ins. No table of maximum calking pitches for a working pressure of 125 lbs. per sq. in. being given, it is necessary to calculate the pitch by equation (92), p. 120,

$$p - d = 21{:}98\sqrt[4]{\frac{t^3}{F}}.$$

For a cover plate $\frac{3}{8}$ in. thick,

$$p - d = 21.98\sqrt[4]{\frac{0.375^3}{125}},$$

$$p = 4.03 \text{ ins.}$$

This pitch is less than the one it is proposed to use. However, the difference is so small (0.03 in.) that the $\frac{7}{8}$ in. rivet selected will be used with an outside cover plate $\frac{3}{8}$ in. thick.

The several laps may now be found from the tables and plots of Chap. III, pp. 137 et seq. The rivets are all $\frac{7}{8}$ in. in diameter and the plate thicknesses as follows:

Shell	$\frac{7}{16}$ in.
Outside cover plate	$\frac{3}{8}$ in.
Inside cover plate	$\frac{5}{8}$ in.
Heads	$\frac{5}{8}$ in.

Ring seam (shell).	Table III.	Lap 1.30 ins.	Use $1\frac{5}{16}$ ins.
Ring seam (heads).	Table III.	Lap 1.16 ins.	Use $1\frac{3}{16}$ ins.
Shell, longitudinal joint.	Table IV.	Lap 1.44 ins.	Use $1\frac{7}{16}$ ins.
Outside cover plate.	Fig. 76.	Lap 1.20 ins.	Use $1\frac{1}{4}$ ins.
Inside cover plate.	Table III.	Lap 1.37 ins.	Use $1\frac{3}{8}$ ins.

Insertion of Longitudinal Joint. — The distance between end ring seams being assumed 24 ft. six short pitches, p_2, Fig. 95, must be subtracted therefrom to determine the number of normal pitches p_1 in the joints. Taking the short pitch eight-tenths that of the normal pitch,

$$p_2 = 0.8 \times 4.06 = 3.25 \text{ ins.}$$

Then $$288 - 6 \times 3.25 = 268.50 \text{ ins.}$$

The number of normal short pitches p_1 is

$$\frac{268.50}{4.06} = 66.25.$$

Use 66 short pitches, or 22 per course.

The actual short pitch is, then,

$$\frac{268.50}{66} = 4.07 \text{ ins.}$$

The final factor of safety may now be found by means of equation (89), p. 117.

$$\text{F. S.} = \left(\frac{8.14 - 0.875}{8.14}\right)\left(\frac{0.4375 \times 55{,}000}{125 \times 30}\right).$$
$$= 5.72.$$

Ring Seam. — The pitch of the rivets in the ring seam joints, as found from Table VII, p. 157, is 2.00 ins. and the efficiency 56.2 per cent.

The number of rivets in a ring seam will be

$$\frac{\underline{\text{Circumference of 60 in. circle}}}{2.00}$$

or

$$\frac{188.5}{2.00} = 94.3 \text{ pitches.}$$

The pitch, as determined from equation (81), p. 115, to allow an efficiency of one-half that of the longitudinal joint, or 44.6, is

$$p = \frac{0.601 \times 45{,}000}{0.446 \times 0.4375 \times 55{,}000}$$
$$= 2.52 \text{ ins.}$$

The number of pitches corresponding is

$$\frac{188.5}{2.52} = 74.7.$$

As a mean try 84 rivets with a pitch of 2.25 ins.

The efficiency of this joint, obtained from equation (79), p. 115, since the rivets will shear, will be

$$V = \frac{0.601 \times 45{,}000}{2.25 \times 0.4375 \times 55{,}000}$$
$$= 0.50.$$

The factor of safety from equation (90), p. 118, is

$$\text{F. S.} = \frac{2 \times 0.50 \times 0.4375 \times 55{,}000}{125 \times 30}$$
$$= 6.42,$$

which is larger than that for the longitudinal joint. The distance between rivet rows on the outside cover plate as found graphically, Fig. 52, or by equation (72), p. 100, is 2.25 inches.

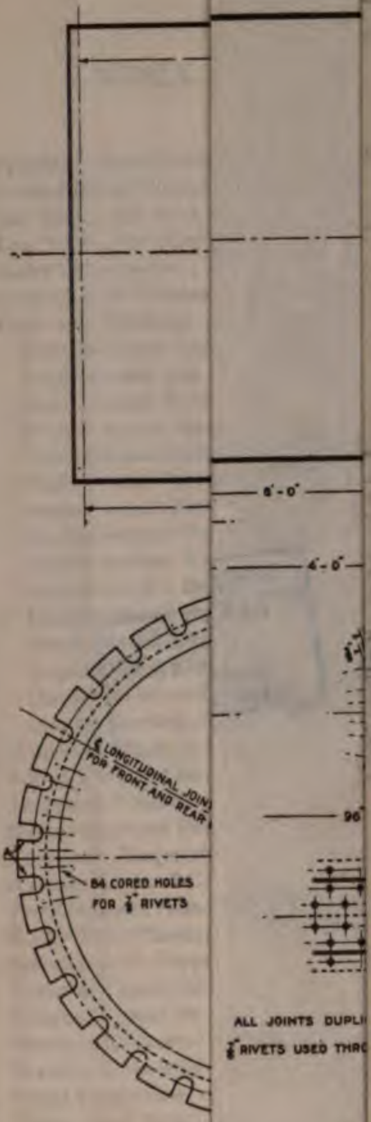
The width of the inside cover plate is

$$2\left(1\tfrac{7}{16} + 2\tfrac{1}{4} + 1\tfrac{1}{4}\right) = 9\tfrac{7}{8} \text{ ins.}$$

The width of the outside cover plate is

$$2\left[1\tfrac{3}{8} + \tfrac{7}{8} + \tfrac{1}{4} + 1\tfrac{1}{4} + 2\tfrac{1}{4} + 1\tfrac{7}{16}\right] = 14\tfrac{7}{8} \text{ ins.}$$

In accordance with the calculations given, the following working drawings, Fig. 197, have been prepared.


6'-0"
4'-0"
96"
℄ LONGITUDINAL JOIN
FOR FRONT AND REAR
84 CORED HOLES
FOR 7/8" RIVETS
ALL JOINTS DUPL
7/8" RIVETS USED THR

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